N71-30391

NASA CR-108577

FINAL REPORT

APOLLO SERVICE PROPULSION SYSTEM ROCKET ENGINE BIPROPELLANT VALVE IMPROVEMENT PROGRAM

Prepared Under
Contract NAS 9-8317

for

Manned Spacecraft Center
National Aeronautics and Space Administration
Houston, Texas

Report NASA CR 108577 February 1971

AEROJET LIQUID ROCKET COMPANY
A DIVISION OF AEROJET-GENERAL
SACRAMENTO, CALIFORNIA

February 1971

Report NASA 108577

APOLLO SERVICE PROPULSION SYSTEM ROCKET ENGINE BIPROPELLANT VALVE IMPROVEMENT PROGRAM

Final Report

Prepared Under
Contract NAS 9-8317

for

MANNED SPACECRAFT CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Houston, Texas

Aerojet Liquid Rocket Company
A Division of Aerojet-General Corporation
Sacramento, California

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF FIGURES (cont.)

FOREWORD

This is the final report of the Apollo Service Propulsion System Rocket Engine Bipropellant Valve Improvement Program. It is submitted in accordance with Exhibit D of Appendix II of Contract NAS 9-8317. The report summarizes all of the work accomplished under this contract. Primary emphasis has been placed upon Phase II of the program which included all work with cryogenic propellants. A more detailed description of the work accomplished during Phase I of the program may be found in the Phase I Interim Report 8317-P1 of July 1970.

The program was administered from the ALRC Apollo Department under the direction of C. E. Teague.

I. INTRODUCTION

Contract NAS 9-8317 started in July 1968. The program objective was to improve the Apollo Service Propulsion System (SPS) bipropellant valve.

The original scope of work consisted of the design, fabrication, and test of one preprototype valve assembly. Primary emphasis was placed upon improved leak rates and assembly procedures. This work was successfully concluded in December 1969.

On 28 June 1970, Contract Modification Order 3S was issued calling for the design and fabrication of two workhorse propellant valves that could be used with the test engine being fabricated on another NASA sponsored contract (NASA 9-8285). These valves were also designed for ease of replacement of poppets and dynamic seals so that material evaluation testing could be accomplished. This effort was concluded in February 1971.

II. SUMMARY

The function of the SPS bipropellant valve is to control the flow of propellants [N_2O_4 and AeroZINE 50 (50/50 UDMH-Hydrazine)] to the engine. Because of the critical nature of the Apollo Mission, the valve was designed with parallel and redundant features which resulted in the valve really being two valves in one assembly.

Each pair of fuel and oxidizer flow passages is controlled by a separate and independent pneumatic actuation system. Each flow passage has two ball valves, each of which has two seals.

The SPS valve has experienced two major problem areas. These were marginal life cycle characteristics, i.e., excessive leakage after cycling, and complicated assembly and repair procedures.

II, Summary (cont.)

The primary objective of Phase I of Contract NAS 9-8317 was to improve upon the undesirable features of the existing valve design. Phase I started during the first week of July 1968.

Related experience was reviewed and tradeoff studies were conducted to establish the basic valve concept. The selected concept retained the basic SPS dual propellant passages with redundant dual seal ball valves, but departed significantly from the SPS in other design areas: (1) the valve assembly was completely modular, i.e., individual, interchangeable ball and seal assemblies, housings, actuators, and actuation systems for ease of maintenance; (2) the ball seals were lifted free of the ball in the first few degrees of motion to reduce the wiping action and improve cycle life; (3) the drive mechanism incorporated idler gears to ensure that the bores of the fuel and oxidizer balls were parallel to the housing bores when in the full open position; and (4) the return spring of the actuators was positioned external to the actuation cavity. The pneumatic actuation control was retained, although it was recognized that an electrical system would offer some advantages.

Following the conceptual design review held at NASA/MSC in early November 1968, the drafting of the detailed engineering drawings was started. First priority was assigned to the valve cartridge, i.e., the module which contains one ball, liftoff cams, and the upstream and downstream seals. As soon as the drawing status permitted a good approximation of the final configuration, a test assembly composed of obsolete SPS components was fabricated to test the feasibility of the concept. The results of these tests showed excellent seal life and unsatisfactory cam cycle life. Consequently, the design was modified to increase the cam bearing width.

At this point, it was decided to depart from the original program plan which had been published at the end of the first program month. This plan

II, Summary (cont.)

called for the Phase I design review and subsequent fabrication of the optimized valve immediately upon completion of the tests conducted with the obsolete SPS hardware. An assessment of the test results indicated that this course of action would be impractical because it had not been possible to maintain the close tolerance required to adequately evaluate the cam and follower assembly. Another unknown was the bellows used in both the upstream and downstream cage assemblies. Consequently, because the cartridge assembly was definitely the most important subassembly and the only one which extended the existing technology, it was advisable to apply extra effort to the component level tests with preprototype cartridge assemblies before committing to the complete valve assembly.

Cartridge dry cycle testing started during the latter part of May 1969. The results of these tests demonstrated the concept to be completely satisfactory.

Assembly of the two-bore valve started in September 1969. This assembly consisted of one fuel bore and one oxidizer bore, i.e., one half of a prototype design. The results indicated the following:

(a) Satisfactory sealing characteristics of liftoff ball seals in which the axial displacement was absorbed by convoluted bellows.

(b) Unsatisfactory characteristics with bending type seals in which no provision was made for axial displacement.

(c) Unsatisfactory characteristics with seals in which the axial displacement was taken out by the toroidal bellows (1 convolute).

II, Summary (cont.)

(d) Liftoff cams and followers performed well in the oxidizer but the lubricant (S-122 Teflon spray) was not satisfactory for fuel service.

(e) The actuators, drive gear, and shaft seal components functioned adequately.

In summary, although some minor problems were yet to be solved, the valve demonstrated a definite potential for improvement over the existing SPS valve design, and further, a cycle life of 20,000 cycles with an attendant maximum leakage rate of approximately 50 cc/hr could be realized.

Testing of the completed valve commenced in November 1969 and terminated in January 1970.

Phase II, the cryogenic propellant portion of the program, started in June 1970. The objective of this program was to support related injector and chamber studies being conducted on another NASA sponsored contract at ALRC (Contract NAS 9-8285).

Two workhorse type valves were designed and fabricated. The valves were designed with a dual objective: (1) to have the capability of being test fired with the injector chamber combination available from NAS 9-8285; and (2) to have enough design flexibility to support laboratory seal testing. Consequently, the valves were overdesigned in that wide allowance was made to provide stock to machine grooves for various types of seals.

The initial study effort was directed toward a bipropellant valve design. However, after a review of potential engine configurations and available bipropellant valve designs, it was decided that a single valve concept would be more appropriate. A single valve offers more versatility with respect to lead lag, mixture ratio control, and engine interface.

II, Summary (cont.)

The preferred valve concept selected was a pneumatically actuated, angled poppet valve. The poppet configuration was chosen because of its high cycle life, minimum seal wear, and adaptability to size requirements. A pneumatic actuation system was selected over hydraulic and electric systems because of intended test stand usage. This actuation method affords a fail-safe capability, ability to vary valve response time without redesign, and to minimize the temperature effect on the actuation media.

Main valve shutoff control is accomplished by a compression molded Kel-F poppet sealing against a 304 stainless seat. The contracting surface of the poppet is a spherical radius and the seat surface is flat. Poppet to seat loading is accomplished by two beryllium copper Belleville spring washers positioned within the actuator piston. The resultant force, 174 lb, is transmitted through the valve shaft shoulder from the inside diameter of the spring washer. Poppet travel or seat penetration, after the valve closing springs have returned the actuator piston to the closed position, is controlled by an adjustable gap between the pintle nut and actuator piston guide. Other pertinent design features are: a readily removable shutoff seat so that other configurations may be evaluated; consideration was given to the possibility of testing different types of shaft seals; the quantity of Belleville spring washers is not fixed but can be varied depending on sealing force requirements.

The primary objective of the test program was to evaluate the cryogenic performance of the polymeric dynamic shaft and piston seals and the poppet seal material. In addition, K_w versus valve position, the ΔP at the design flowrate, and minimum achievable poppet travel time were determined.

A summary of tests conducted and the associated test pressures is as follows:

II, Summary (cont.)

Test	Leak Check Pressure, psig 10	30	100	200	500	700
Acceptance leak check	x	x	x	x	x	x
Post 10, 50, 75, and 100 dry cycles leak check	x	x				x
Post 100 wet cycles leak check		x				x
Post 500 wet cycles leak check		x				x
Post 1000 wet cycles leak check		x				x
Post 5000 wet cycles leak check		x				x
Post 7000 wet cycles leak check		x				x
Post 10,000 wet cycles leak check		x				x
Post 10,000 cycles at ambient leak check		x				x

The valve configuration for the first test series of 10,000 cycles was with RACO shaft and piston seals and the Kel-F molded poppet. An additional 10,000 cycle test was conducted with Delta dynamic seals and a partially encapsulated Teflon poppet. Primary design criteria and min/max leakage data from the test series are as follows:

Leakage Rate, SCC/HR GHe

	30 Psig		700 Psig	
	Min	Max	Min	Max
Actuator Piston Seals				
RACO	0	8,010	6,888	882,000
Delta	0	30	0	48,000
Rear Shaft Seal				
RACO	0	2,244	960	184,000
Delta	0	125	0	14,400

II, Summary (cont.)

	30 Psig Min	30 Psig Max	700 Psig Min	700 Psig Max
Front Shaft Seal (Toward Poppet)				
RACO	0	12,000	0	96,000
Delta	0	45	90	4,002
Poppet				
Kel-F	0	0	0	0
Teflon	0	90,000	0	10,410

NOTE: The design criteria for the RACO and Delta dynamic seals are 100 scc/hr GHe, and 100 scc/hr GHe, respectively, past the poppet. The above data are the minimum and maximum leakage rates noted throughout each of the 10,000 LN_2 cycle tests.

Water flow testing was performed to determine the valve resistance (K_w) and pressure differential at the designed flowrate. Valve resistance ($K_w = \frac{\dot{w}}{\sqrt{\Delta P \times S.G.}}$) was calculated at 10% increments of valve opening. At 100% open the valve resistance was 3.72. The pressure drop across the valve was 16 psi at a water flowrate of 16 lb/sec and at a supply pressure of 700 psig. Minimum valve poppet travel time tests were accomplished using a Marotta solenoid valve (orifice size = 0.190 in.) attached directly to the OME valve actuator inlet port. The fastest poppet travel time achieved was 0.007 sec opening and 0.035 sec closing with a supply pressure of 800 psig (GHe).

A minimum cycle life of 10,000 cycles at liquid nitrogen temperature was desired with leakage rates within the design criteria. The Kel-F poppet configuration achieved this goal; however, neither the RACO or Delta dynamic seals remained within the design limits at low temperature. During the

II, Summary (cont.)

low-pressure leakage tests, 30 psig GHe, the maximum Delta seal leakage was 125 scc/hr as compared to 2,244 scc/hr with a RACO seal in the same location. The maximum Delta seal leakage at the higher test pressure, 700 psig GHe, was 48,000 scc/hr as compared to 882,000 scc/hr with a RACO seal. Dynamic seal leakage, in the RACO and Delta configurations, is attributed to seal shrinkage away from the sealing surface. This is confirmed by the leakage rates being within limits at ambient temperature both before the start of the 10,000 LN_2 cycles and at the conclusion. Indications are that the Delta dynamic seals are superior to RACO seals in cryogenic temperature applications; however, further effort would be required to verify that the seals could be brought to within acceptable leakage limitations.

III. TECHNICAL DISCUSSION

The various elements or tasks of the Valve Improvement Program (VIP), which resulted in the fabrication and test of an optimized SPS valve concept, are reported in detail in the Phase I Report. That report, AGC 8317-P1, dated July 1970, is concerned only with the hypergolic propellant phase of the contract. This report contains information on both the hypergolic and cryogenic propellant portions of the program with emphasis placed upon the cryogenic propellant portion.

A. HYPERGOLIC PROPELLANT PHASE

1. Program Tasks

As stated previously, the objective of the hypergolic portion of the contract was to improve upon the existing SPS engine valve. The approach involved a review of previous related experience, design selection based upon the results of that review, feasibility tests of critical components of the selected design, and finally, fabrication and test of one preprototype unit.

Because this report is concerned primarily with the cryogenic propellant phase of the program, only representative data are included in this the hypergolic propellant portion. Also, the tasks are reported as independent and distinct events although there was considerable overlapping both in technical make-up and in schedule.

a. Review of Previous Experience and Tradeoff Studies

A comprehensive review of related experience within Aerojet was initiated in the second contract week. The purpose of this review was to identify past problems so that appropriate countermeasures could be instituted early in the program.

III, A, Hypergolic Propellant Phase (cont.)

In addition to sources within Aerojet, technical discussions were held with two of the Apollo subcontractors who also were using ball-type propellant valves.

On 23 September 1968, a technical exchange meeting was held at Thompson Ramo Wooldridge (TRW, Inc.), the manufacturer of the Lunar Descent Engine bipropellant shutoff valve. Much of the discussion involved TRW's experience with various materials and coatings in the N_2O_4 and AeroZINE 50 environment.

Bell Aerosystems experience with the Lunar Module Ascent Engine bipropellant valve was reviewed at Niagara Falls, New York, 18 October 1968.

A review of the various design features of this valve showed that although two seals are used on each ball [also being considered for the Valve Improvement Program (VIP) valve], the design concept was quite different. The ball was allowed to float between the Teflon seals. The main seal is the downstream seal which is fixed in place. The upstream seal is not really a seal in that it does not seal between seal and the body - its primary purpose being to transmit load from Belleville springs through the ball and against the downstream seal. The review of previous experience was concluded at this point with the results applied to the tradeoff studies.

Tradeoff studies were also initiated in the second contract week and conducted in parallel with the review of previous experience. The purpose of these studies was to select the optimum design from the various design concepts. As a part of this study, formal review meetings were held at Aerojet which were attended by personnel from Engineering, Test, Quality Control, and Apollo departments which had specialized applicable knowledge and experience.

III, A, Hypergolic Propellant Phase (cont.)

The first of these reviews was held during the second program month and resulted in the establishment of design ground rules which would serve to narrow the field of potential designs to receive detailed studies. These were:

(1) The improved valve should be physically and functionally interchangeable with the existing valve. It was a design goal to attain a leakage rate of less than 10 cc/hr of helium, although this goal was recognized as arbitrary.

(2) Weight was important but not an overriding factor.

(3) The valve should have improved reliability and fabricability. A design goal was to reduce fabrication costs to one-third that of the existing valve.

(4) The complete development of an electrical actuation system was beyond the scope of the present contract. However, consideration should be given to tailoring the design such that electrical actuators could be used on the new valve without requiring changes to the valve assemblies other than replacing the pneumatic actuators.

(5) An improved SPS pneumatic actuation system should be used.

(6) The existing engine and injector interfaces should not be changed except as required to avoid severely compromising the bipropellant valve design.

(7) The optimized bipropellant valve should be functionally interchangeable with the existing SPS valve.

III, A, Hypergolic Propellant Phase (cont.)

(8) The valve would use cam-lifted ball seals to minimize seal wear.

(9) Cartridge modules, installed in quadmonopropellant valves would be used. The cartridge module would contain, as a minimum, a ball and two seals, with the capability of leak testing the seals as a subassembly. The cartridges, if not alike, would be noninterchangeable.

(10) The valve housing would permit a cartridge to be installed without requiring disassembly of another cartridge. It was also a design goal to permit replacement of a valve cartridge without disassembly of the valve halves, or inboard shaft seals.

(11) The actuator housing would be separate from either valve half. The valve would be identical, except that the fuel and oxidizer valve-to-actuator interface would be noninterchangeable. The interchangeability between electrical and pneumatic actuation systems would be a consideration in the design of the actuator housing.

(12) It was a design goal to provide adjustable valve end position stops.

(13) Bipropellant valve assembly handling provisions should be incorporated in the design.

Once the basic groundrules had been established, work on numerous specific design concepts was started.

The matrix listing the advantages and disadvantages of each of the concepts considered is included in Figure 1.

III, A, Hypergolic Propellant Phase (cont.)

Upon completion of the tradeoff studies, another internal Aerojet design review was held that resulted in the selection of the basic design.

b. Design Selection and Description

The results of the foregoing studies are embodied in the valve design shown in the cutaway diagram (Figure 2). The salient features of the design are:

(1) Housing

The valve housing consists of two identical castings. Each casting has two bores which form parallel propellant flow passages (in Figure 2, the two bores on the left are within one casting). The bores are machined to accept cartridge assemblies from each end. These assemblies are retained in position by coverplates, which are bolted over the ends of the flow passages after assembly. In addition, the housing contains keyed shafts which transmit opening and closing torques from the actuation gearbox to the valve balls. A shaft and key are shown in the upper bore of Figure 2 where the cartridge assembly has been omitted for clarity. The key is shown in the vertical position. It was, of course, rotated to the horizontal attitude during cartridge assembly. Drain, bleed, and test ports were also included in the housing assembly. These ports are shown in Figure 2 (left side).

(2) Actuator

Four actuator assemblies are included in the bipropellant valve assembly. Each actuator controls one fuel and one oxidizer ball seal assembly. The actuators are shown in the assembled condition bolted between the housings in Figure 2. The linear motion of the pneumatically

III, A, Hypergolic Propellant Phase (cont.)

actuated piston is converted to rotary motion by a rack and pinion. This motion is then transmitted to the ball shafts (described in the housing discussion above) through a gear train. The gear ratio of the gear train provides for an oxidizer lead upon opening which is required to ensure a smooth engine start transient.

The actuator is spring closed with the springs located outside of the pressurization cavity to preclude contamination of the pneumatic system by spring-generated particulate matter.

(3) Cartridge Assembly

The cartridge assembly shown in Figure 3 contains the seals, and because of this, it is the most important of the subassemblies. In operation, the key in the drive shaft (see Figure 2) engages the slot or keyway in the cam. The rotary motion imparted by the drive shaft through the keyway to the cam causes the cam follower to ride up on the cam lobe. The motion of the cam followers is then transmitted to the seal lift ring, which in turn lifts the seal away from the surface of the ball. Seal motion is taken by a bellows on the downstream seal and by bending of the seal in the upstream seal. Seal loading was accomplished by coil springs located around the circumference of both the upstream and downstream seals. The seals were lifted off the sealing surface in the first 6 degrees of ball motion, thereby minimizing the possibility of seal damage by rubbing on the ball surface.

(4) Actuation System

The SPS-type pneumatic actuation system was retained in the interest of economy although it was repackaged as shown in Figure 4.

III, A, Hypergolic Propellant Phase (cont.)

c. Single-Position Valve Tests

The original purpose of the single-position tests was to support the design effort and provide a quick, economical method of evaluating design concepts before they were incorporated in the final design. The first tests were to be conducted with surplus or expended hardware from the SPS program and subsequent tests were to be conducted with breadboard versions of VIP hardware.

The single-position tests were later expanded to include tests with prototype cartridge assemblies. This change was necessary because the cartridge assembly was completely different from the basic SPS design, and as a consequence, it was not possible to evaluate the design to the extent that it would be prudent to define the total valve after only limited testing with used hardware. The tests conducted with both types of hardware are summarized in the following paragraphs:

(1) Single-Position Tests with SPS Hardware

The single-position testing was initiated in mid-November 1968. The primary purpose of the first series of tests was to gain insight into the operation of the cam. The secondary objective was to obtain preliminary data to verify that the reduced rubbing distance of the seal would produce a significant reduction in seal wear and improvement in leakage.

The hardware required for performing these tests consisted principally of the existing Apollo ball valve hardware with some minor modifications which were: (1) the ball was made narrower so that the ball and cams would fit into the existing cage; (2) the cage was shortened to accommodate the seal lift ring; and (3) the shaft was slotted to key the outboard cam to the shaft. New parts, such as the cams, followers, and the seal lifting mechanisms, were designed and procured specifically for these tests. The cam design was recognized as being one of the most challenging aspects of the new components because of the long cycle life required at high loads. Figure 5 is a drawing of the single position tester, and Figure 6 is a photograph of the components.

The first tests were conducted with liftoff cams and followers fabricated from type 17-4 stainless steel and hard-coated with a flame dispersion identified by the trade name of Colmonoy No. 6. Succeeding tests were conducted with Colmonoy No. 72. The tests indicated that while the hard coating process was probably feasible, it was not desirable for this program because of the difficulty of maintaining the desired product control. Consequently, all subsequent effort was devoted to cam designs which did not require the use of coatings to obtain the desired surface hardness, with primary emphasis placed upon obtaining suitable cam material to use with a Haynes Stellite 6B follower.

Cycle testing of the seal was complicated by the use of the hard-coated cam because of the contamination generated by the hard coating. However, one unit went through 10,000 N_2O_4 cycles with no apparent wear on the seal. The lack of wear on the seal surface after 10,000 cycles proved that the theory of lifting the seal away from the ball provided a greatly extended seal life which was the objective of the test series.

(2) Single-Position Tests with Prototype Hardware

(a) Oxidizer Cartridge Tests

Sufficient hardware was received 21 May 1969, to allow the start of the single-position cartridge testing with prototype hardware. The initial work consisted of checking out the assembly shop aids

III, A, Hypergolic Propellant Phase (cont.)

and the assembly procedures. The detail parts for a cartridge are shown in Figure 7. The first cartridge assembly was prepared 27 May 1969. The assembled cartridge is shown in Figures 8 and 9.

The first cycle tests were conducted with the cams and followers dry, followed by tests with the parts lubricated with FS-1281 and S-122. Cycling with the dry, unlubricated cams and followers resulted in noisy, squeaky operation, a rapid increase in shaft torque, the generation of black deposits, and scratched hardware. Cams and followers lubricated with FS-1281 operated smoothly with no noise and low torque. After 500 cycles, operation was still smooth with a moderate cam breakaway torque increase (from 43 to 55 in.-lb). Because FS-1281 tends to wash out in propellant and thick coatings will decompose and flake off, the FS-1281 was replaced by S-122 spray lubricant for all subsequent tests using Stellite 6B and Stellite 19 cams and followers.

The oxidizer cartridge was then assembled with Stellite 19 cams and Stellite 6B followers, both lubricated with FS-1281. Special tooling was fabricated to facilitate assembly. Wet cycle testing was initiated in the components evaluation laboratory on 12 June 1969. The torque and leakage results are tabulated in Figures 10 and 11.

It will be noted that the test was interrupted at 1500 cycles due to an increase in torque (see Figure 10). Cam galling was suspected as the cause of the torque increase; however, upon disassembly, the cams were found to be in good condition. The cams were relapped and testing resumed. Torque again gradually increased to a relatively constant value. These data, which were subsequently confirmed on succeeding tests, indicated that the cam and followers must be "run in" before reasonably constant torque values can be obtained.

III, A, Hypergolic Propellant Phase (cont.)

The leakage and cycle life characteristics shown in Figure 11 of 250 cc/hr (maximum) in 20,000 cycles represented a significant improvement over the production valve.

(b) Fuel Cartridge Tests

A fuel cartridge was also assembled and tested. The purpose of the tests was to demonstrate that dry cycling could be accomplished prior to wet cycling as had occurred with the N_2O_4 assembly and also to evaluate the effect of the increased wiping action associated with the fuel cam (9 degrees versus 6 degrees for the oxidizer cam). The results of these tests are shown in Figures 12 and 13. It will be noted that the results were not as good as those obtained with the oxidizer cartridge but were still significantly better than had been possible with a nonlifting ball seal.

Following dry cycling, water flow tests including K_w and seal bite were conducted; no anomalies were disclosed.

Cartridge testing was terminated at this point because the test results had demonstrated conclusively the improved potential of the seal liftoff cartridge assembly and readiness to commit to full-scale valve testing.

d. Optimized Design

This task consisted of the design and drafting effort associated with the actual making of engineering drawings, specifications, etc. Seventy engineering drawings were completed. Also included in this task category, were the friction and wear tests of various cam and follower materials which were conducted in the materials laboratory to back up the single-position tests. The test setup could not duplicate the cam and follower conditions

III, A, Hypergolic Propellant Phase (cont.)

closely enough to give quantitative data on cycle life but it did provide comparative data to indicate which material combination should be the best. The combinations tested and their comparative ratings are as follows:

Specimen Materials			
Cam	Follower	Rating	Remarks
Colmonoy 72	Stellite 6B	Comparison Base	Produced black powder same as actual cam tests
Stellite 6B	Tungsten Carbide*	Fair	---
Stellite 6B	Stellite 19	Fair	---
Stellite 6B	Aluminum Oxide** (99.5%)	Fair	---
Stellite 6B	Aluminum Oxide*** (99.9%)	Better	---
Stellite 19	Tungsten Carbide	Superior	---

* Carboloy 320, a cemented carbide (General Electric Company).
** Wearbox 995, a cemented aluminum oxide (Western Gold & Platinum Company).
***Carboloy 0-30, a cemented aluminum oxide (General Electric Company).

The tests showed that the best results would be expected using cams of Stellite 19 and a follower with tungsten carbide contact surfaces bonded in place. Rework of existing Stellite 6B followers to braze tungsten carbide inserts to the tips was completed; however, these followers were never tested because of the success experienced with the Stellite 19 cam and Stellite 6B follower in the cartridge tests.

e. Test Plan

Three test plans were prepared and submitted: one for the single-position tests with SPS hardware, one for the cartridge tests, and one for the two-bore valve assembly, which is included in the Appendix.

III, A, Hypergolic Propellant Phase (cont.)

f. Valve Timing Rig Studies

The original purpose of these studies was to investigate the feasibility of a device for timing the bipropellant valve actuation system which would provide a substantial reduction in the total number of valve ball cycles during buildup. Because of the excellent cycle life capability demonstrated by the cam lift seal, this device is no longer needed; consequently, no effort was expended on this task.

g. Design Guide

A draft copy of a design guide was submitted in June 1969. The comments of the NASA Technical Monitor were received in July 1969 and were incorporated.

h. Two-Bore Valve Tests

(1) Assembly

The "two-bore" valve tests were so named because the tests were conducted with one-half of the prototype valve, i.e., one fuel and one oxidizer bore. A photograph of the valve during test is shown in Figure 14. This test configuration was selected for reasons of economy because the single-position tests had been more extensive than originally planned, and as a consequence, less funding was available for the final test series. Despite this change, little or no impact was imposed upon the technical objectives because the components were selected with the objective of obtaining information on the most promising candidates resulting from the single-position tests, as shown in the following table:

III, A, Hypergolic Propellant Phase (cont.)

Position	Cam Material	Follower Material	Seal Type	Bellows Type
Fuel				
Upstream	Stellite 19	Stellite 19	No. 1 - Liftoff No. 2 - Liftoff	Toroidal (Figure 15) Convoluted (Figure 16)
Downstream	Stellite 19	Stellite 6B	No. 3 - Bending No. 4 - Liftoff	Convoluted Convoluted
Oxidizer				
Upstream	Stellite 19	Stellite 6B	No. 1 - Liftoff No. 2 - Liftoff	Toroidal Convoluted
Downstream	Stellite 19	Stellite 19	No. 3 - Bending No. 4 - Liftoff	Convoluted Convoluted

(2) Test

Testing started in November 1969 with 500 dry cycles. Problems were encountered in the fuel circuit due to removal of the Teflon lubricant during a pretest cleaning with alcohol.

Wet cycle testing started in December 1969. At the completion of the first 250 cycles, the fuel half had excessive leakage and was removed from the oxidizer valve for failure analysis.

The following conditions were noted upon teardown of the fuel valve: all cams and followers were galled. Galling occurred in the center portions of the loading surface, and the lubricant had been removed from the loading surfaces of the cams and followers.

Rework of the fuel half consisted of the following: cams and followers were relapped or replaced as required. A Microseal 100-1 coating was applied to all cams and followers to reduce the coefficient of

III, A, Hypergolic Propellant Phase (cont.)

friction between cam and follower. Glass-filled Teflon ball seals were installed in the No. 1 and 2 positions.

During the fuel valve rework, an additional 500 wet cycles were conducted on the oxidizer half.

The fuel valve was reinstalled on the oxidizer half, and testing resumed with an additional 500 wet cycles on oxidizer and 500 dry cycles on the fuel valve.

Additional wet cycling was completed comprising a total of 1500 wet cycles on the oxidizer half and 500 wet cycles on the fuel half. Post-cycle check indicated excessive leakage past the No. 3 and 4 fuel ball seals and also past the No. 3 oxidizer seal.

Post-cycle leakage, functional, timing and seal backpressure relief data are shown in sequence in Figures 17 through 21.

(3) Posttest Teardown

The following conditions were observed during post-wet-cycle teardown and evaluation.

Fuel Cams and Followers - All parts showed signs of severe wear and galling in the centers as shown in Figure 22. There was no indication of lubricant on the loading surfaces. The Microseal was removed from the worn areas.

Oxidizer Cams and Followers - Minor wear similar to that shown in Figure 23 was noted on all parts.

III, A, Hypergolic Propellant Phase (cont.)

Some lubricant (very little) remained on the loading surfaces. The minor wear was in the centers of the cams and followers.

Fuel Ball Seals

No. 1 - No visible damage

No. 2 - Small piece of metal on sealing surface (probably from cams and followers)

No. 3 - Minor scratches and small metal chips on sealing surface

No. 4 - Minor scratches

Oxidizer Ball Seals

No. 1 - Minor scratches

No. 2 - Minor scratches

No. 3 - No apparent damage

No. 4 - Three deep scratches on sealing surfaces

There was no visible indication of wear or damage on the actuator components, drive gears, shafts, or shaft seal components.

2. Conclusions and Recommendations

The basic conclusion of the effort is that the modular bipropellant valve assembly concept with cam-lifted seals is definitely feasible. The concept of the modular cam-lifted seal design is considered adequately demonstrated and any further effort would be of a design refinement category. The two-bore valve actuators and associated gearing performed satisfactorily during testing. Those areas which would require further development to meet all the design objectives are development of the fuel cams and cam followers to prevent galling in AeroZINE 50 and development of an improved upstream seal design and/or use of a filled Teflon seal material to minimize seal flaking.

The basic conclusion is substantiated by several subordinate conclusions in the following areas:

a. Sealing

(1) The lifted seal concept results in a significant improvement in both life and sealing capability over the SPS rubbing seal design because of its reduced contact time and increased seating loads.

(2) Axial seal displacement is preferable to the bending seal concept because the load on the cam is reduced and the seal is not deformed. Some permanent deformation and resultant loss in sealing characteristics probably occurs due to bending. The No. 3 seal, bending type, did not perform satisfactorily during two-bore valve testing. Test results demonstrated superior performance of downstream seals. Development of an improved upstream seal through the use of a redesigned bellows or filled Teflon seal material is one of the two primary outstanding design refinements. Filled Teflon seals that would be considered for evaluation would include calcium fluoride, boron nitride, carbon and graphite, and borosilicate glass.

(3) Dual ball seals add to reliability, but also add complexity and cost. Tradeoff studies would be required for specific application before committing to either the single or redundant seal concept.

(4) Additional effort is required to establish run-in procedures. Tests with both the oxidizer and fuel cartridges showed a reduction in leak rate after the initial cycling, leading to the conclusion that

III, A, Hypergolic Propellant Phase (cont.)

run-in cycles should be accomplished as a part of the assembly procedure prior to acceptance test. However, the run-in cycling accomplished on the two-bore valve assembly did not provide the desired results, indicating further development of the run-in procedures is necessary. Run-in cycle rate is critical. The leakage results obtained with the oxidizer cartridge assemblies when compared with those obtained with fuel cartridge indicate that the slightly longer rubbing interval, e.g., 6 degree oxidizer versus 11 degree fuel, results in a marked decrease in the number of cycles before Teflon builds up on the ball and excessive leakage is experienced. Based on the single-position test results, the ALRC Material Department recommended a reduced cycle rate from the 6 cycles/min to approximately 2 cycles/min to reduce the temperature buildup which is hypothesized as the cause of the premature failure of the oxidizer seals after only 2000 dry cycles. Opening and closing rates remained unchanged at approximately 0.5 sec throughout the program.

(5) Initial run-in of cams is critical and a light film of Teflon is beneficial while the cams and followers are seating in. The run-in methods used to date do not provide sufficient contact between the cams and cam followers and, in the case of fuel valve cartridges, this reduced contact area leads to excessive cam and cam follower galling during valve cycling with propellants. The combination of a Stellite 19 cam and Stellite 19 cam follower appears to be the most resistant to wear and galling during valve cycling. However, further investigation must be made into the feasibility of lapping the cam to the cam follower, resulting in matched set usage. Assuring adequate cam contact area is available and development of a lubricant or antigalling material remains as a desired design refinement for AeroZINE 50 service.

b. Assembly

Assembly procedures must be refined. An excessive number of builds have been required prior to obtaining required leak rates. This is a typical development problem which can only be solved by increased familiarity with the hardware and the evolution of improved assembly methods and tooling.

III, Technical Discussion (cont.)

B. CRYOGENIC PROPELLANT PHASE

1. Design Selection

The valve design, shown in Figure 24, was designed to satisfy criteria indicated in Figure 25. These criteria represent a compromise between anticipated ground and flight test requirements. Flight applications will require the valve to flow liquid hydrogen or oxygen and ground testing will utilize gaseous propellants. The high valve inlet pressure of 1600 psi reflects engine test facility requirements. In flight, the inlet pressure condition would be approximately 700 psig. This condition results in a trade-off with respect to valve size and pressure drop. Sizing for the valve design was defined by the required LH_2 and LO_2 flowrate of 2.52 lb/sec and 15.08 lb/sec, respectively, with a pressure drop of 25 psi. The thrust level of the engine design is 8000 lb. The 25-psi pressure drop, when flowing liquid, is equivalent to a 200 to 300 psi differential when using gaseous hydrogen and oxygen. The equivalent orifice size of the valve required was determined to be 1.25 in.

Because of valve size, design simplicity, high cycle life, and excellent sealing characteristics the poppet was selected as the shutoff element. The poppet shaft and actuator piston dynamic seals chosen were a RACO Teflon-jacketed spring-loaded seal (RACO basic PN 10062 and 10066). This type of seal uses a U-shaped spring encased with a U-shaped Teflon jacket. The spring is designed to apply continuous load and eliminate the cold flow and dimensional instability problems associated with Teflon. There are no gaps in the load applied to the jacket as are encountered with other Teflon seals using a garter-type spring. Another more conventional choice of a sealing method would have been a metal bellows. This approach was rejected because one of the objectives of this program was to determine if an acceptable substitute to a bellows was available.

III, B, Cryogenic Propellant Phase (cont.)

The main shutoff seal configuration selected for the primary design was a Kel-F compression-molded poppet on a stainless pintle. This method of securing a plastic poppet to its pintle has been successfully used on other valve programs, i.e., Titan I and Post-Boost Propulsion Systems.

The three basic selections for an actuation system are hydraulic, electric, and pneumatic. For purposes of this program and giving weight to test stand application, a pneumatically actuated piston with return closing springs was selected. This method affords a fail-safe capability, ability to vary valve response time without redesign, and minimizes the temperature effect on the actuation media. A photograph of the components comprising the actuation system is included in Figure 26.

Shutoff control and sealing is achieved by the molded poppet contacting an angled stainless steel seat. The seat is a body insert so that different configurations could be tested. Poppet-to-seat load, 174 lb, is controlled by two Belleville spring washers contained within the actuator piston. The seat load is independent of the valve closing spring force. Poppet-to-seat contact travel is controlled by a gap between the shaft nut and spring guide. This is shown in Figure 24.

The poppet guide was designed to provide uniform distribution of forces resulting from the supply pressure and flow media. This was accomplished by placing three evenly spaced 1.00-in.-dia holes in the poppet guide, none of which was aligned with the valve inlet port (see Figure 24). Valve poppet stroke was designed to be 0.500 in. and is controlled by the distance between the spring guide and the actuator cover shaft guide (see Figure 24). The stroke may be increased by reducing the length of the cover shaft guide.

III, B, Cryogenic Propellant Phase (cont.)

During an opening sequence, actuation pressure is admitted through the actuator inlet port. The required pressure to start motion is between 250 and 300 psig. The piston and spring guide move until the guide comes in contact with the actuator cover. Poppet motion starts when the gap between the shaft nut and spring guide is taken up. In closing, the piston cavity is vented until the closing spring force is greater than the pressure force that holds the valve open. The springs then push the spring guide and piston until the piston comes in contact with the valve body. Final poppet/seat loading is then applied by the Belleville washers acting on the poppet shaft shoulder.

At the completion of the initial 10,000 liquid nitrogen cycles a brief redesign of the valve poppet and shaft/piston dynamic seals was initiated. It was decided that an additional poppet material would be evaluated and the configuration of the dynamic seals would be changed from RACO to Delta. A poppet assembly was modified to contain a Teflon insert which was secured and loaded in position by a bolt attached to the pintle. This is shown in Figure 27. The valve body, piston, and spring guide were modified and additional ports made to allow use of the Delta dynamic seals. The Delta seal configuration used is a graphite-filled Teflon seal ring which is loaded both axially and radially by a wedge ring. The loading force is a result of helical springs placed around the circumference of the wedge ring. A cross-section of a typical Delta seal installation is shown in Figure 28.

A listing of the source of the valve components is shown in Figure 29.

III, B, Cryogenic Propellant Phase (cont.)

2. Procurement and Fabrication

Components for assembling two complete valves plus spares were procured. Pilot valves, considered adequate for test purposes, were available. The redesign effort, at the midpoint of the test program, required that Delta seal assemblies and spare seals be purchased.

3. Assembly

Assembly of one valve for testing started during the first week of December 1970. Photographs of the required components are shown in Figure 26. Upon completion of the initial 10,000 cycles the body, piston, and spring guide of the other valve, not previously assembled, were modified for the Delta seal configuration.

4. Testing

The component test program was initially directed to demonstration of the valve functional and flow capabilities prior to use for engine testing. The test plan consisted of a proof, leak, functional, 100 dry cycles, 10,000 liquid nitrogen cycles, hydraulic K_w determination, and minimum response testing. In addition, the 10,000 liquid nitrogen cycle test would be repeated to evaluate a different poppet material and Delta dynamic shaft/piston seals. The detailed test plan is shown in the appendix.

Major categories comprising the test program are detailed as follows:

III, B, Cryogenic Propellant Phase (cont.)

a. Dry Cycling

Upon completion of the initial proof and leak checks, the valve assembly (RACO dynamic seals and a Kel-F poppet) was subjected to 100 dry cycles with 30 psig supplied to the valve inlet port. Valve response time was established at 0.150 $\pm$ 0.030 sec opening and 0.100 $\pm$ 0.040 sec closing by orificing the pilot valve. The purpose of the cycling was to determine if there was any degradation of poppet and dynamic seals. Leak checks of the poppet and dynamic seals were conducted at the completion of 10, 50, 75, and 100 cycles. Test pressures were 10, 30, and 700 psig GHe. The front main shaft seal and actuator piston seals indicated leakage at 700 psig and at all test points of the 100 cycles. The leakage trend was decreasing for each seal. The front shaft seal leaked 90 cc/10 min GHe at the end of 10 cycles and 60 cc/10 min at 100 cycles. The piston seals leaked 8 cc/10 min at the conclusion of the first 10 cycles and 3.5 cc/10 min at 100 cycles. This is considered to be a normal pattern in some valve applications because of a wearing-in process of the seal. Data are presented in Figure 30. This test was not repeated with the Delta seals or the Teflon poppet.

b. Low-Temperature Cycling

(1) RACO Seals

This test phase was conducted in two parts, the first being with RACO dynamic seals and a Kel-F poppet and the second with Delta seals and a Teflon poppet. Low-temperature cycling was accomplished by applying 700-psig LN_2 (approximately -290°F) to the valve inlet and cycling it 10,000 times at the required response time. A hand valve downstream of the OME valve regulated the LN_2 discharge. A photograph of the test setup is

III, B, Cryogenic Propellant Phase (cont.)

shown in Figure 31. Leak checks of the poppet, dynamic, and static seals, at 30 and 700 psig GHe, were conducted at the conclusion of 100, 500, 1000, 3000, 5000, 7000, and 10,000 cycles. The poppet did not leak during any of the post-cycle leak checks. All of the dynamic shaft and piston seals indicated leakage at each test point. A trend in leakage rates for the dynamic seals was not established. Rates were erratic and probably the result of temperature changes within the valve assembly due to time variation between end of cycling and start of leak checks. The front main shaft seal leakage ranged from 0 cc/hr GHe to 20,000 cc/hr, at 30 psig, throughout the 10,000 cycles. The rear shaft seal leakage was from 0 cc/hr GHe to 3740 cc/hr at 30 psig. Piston seals ranged from 0 cc/hr GHe to 13,350 cc/hr at 30 psig. Data at all test points are presented in Figure 32.

The dynamic seal leakage is attributed to seal shrinkage away from its sealing surface. This is substantiated because the leakage rates returned to within acceptable limits during post 10,000 cycles ambient temperature leak checks.

A manual functional test was conducted intermittently throughout the cycling program. The purpose of this test was to determine changes in actuator force requirements which would indicate friction variations. Pressure to open and close the valve was accomplished manually with the pilot valve open. The pressure point at which the valve would start motion and achieve full open was observed visually and from oscillograph records.

At the end of 5720 LN_2 cycles, the potentiometer oscillograph trace was noted to be erratic with indications that the valve was not going to open. Testing was terminated and the valve partially disassembled for inspection. There was no apparent evidence of galling, binding, or residual

III, B, Cryogenic Propellant Phase (cont.)

moisture from ice which may have prevented the valve from opening. The potentiometer was then checked at low temperature and it was found to become erratic at -50°F and progressively worse with decreasing temperature. It was not determined whether this potentiometer or the design was inadequate. To continue testing, the potentiometer was mounted outside of the actuator cover on an extension tube. The potentiometer was connected to the valve shaft by an extension. A Teflon insulator was also placed between the actuator cover and the extension tube to limit heat transfer. Testing continued without further problems.

(2) Delta Seals

The 10,000 LN_2 cycles were repeated using Delta dynamic seals and a partially encapsulated Teflon poppet. This cycling test was conducted in the same manner as the initial 10,000 cycles. The poppet leakage at the end of the first 100 LN_2 cycles was 800 cc/hr GHe at 30 psig and 150 cc/hr at 700 psig. Leakage returned to zero during the post 500 and 1000 cycle checks. At the conclusion of 5000 cycles, the leakage was 150,000 cc/hr at 30 psig and 17,350 cc/hr at 700 psig. The leakage returned to zero at the end of 7000 cycles and then leaked 800 cc/hr at 30 psig at the conclusion of 10,000 cycles. Leakage at 700 psig, after 10,000 cycles, was zero. At ambient temperature, there was no leakage past the poppet. There was one instance of poppet leakage which was attributed to contamination, probably of a non-recurring nature.

Delta seal leakage rates, during the second 10,000 cycle series, were significantly less than the dynamic RACO seals. The piston seal leakage rate, 80,000 cc/hr GHe at 700 psig noted during post 500 cycle tests, was the most severe leakage encountered. During post 10,000 cycle ambient leak tests, there was no leakage past any of the Delta seals. Leak test results are shown in Figure 33.

III, B, Cryogenic Propellant Phase (cont.)

The improved sealing characteristics of the Delta seal as compared to the RACO seal are due to a harder seal material, graphite-filled Teflon, and more uniform radial loading. Most significant is the wedge ring of the Delta seal assembly. A typical Delta seal installation is shown in Figure 28. This ring applies the axial and radial forces to the seal ring resulting from helical springs around the circumference of the wedge ring. Depending on seal usage, temperature, pressure, and media, the wedge angle may be changed to fit the configuration. The seal assemblies used on this program were procured with a nominal 45 degree wedge angle. This angle may be increased to 56 degrees with a resultant increase to the radial force component and a corresponding decrease in the axial component. Test data indicate less leakage with Delta seals and this leakage is also the result of seal shrinkage away from the sealing surface. This is confirmed by the leakage rate returning to zero during ambient checks. If the wedge angle had been greater than 45 degrees, then the leakage rate should have improved accordingly.

c. Water Flow Tests

Water flow tests were conducted to ascertain that the valve meets the design criteria, and to determine the hydraulic K_w.

$$K_w = \frac{\dot{w}}{\sqrt{\Delta P \cdot S.G.}}$$

Valve resistance (K_w) was determined at various increments of valve opening with a full open value of 3.72. With a ΔP of 16 psi the flowrate was 16 lb/sec. Plots of K_w versus percent of valve opening and ΔP versus flowrate are presented in Figures 34 and 35.

III, B, Cryogenic Propellant Phase (cont.)

d. Minimum Response Testing

Minimum response tests were conducted to determine the fastest valve poppet travel both moving open and closed. The fastest times achieved were 0.007 sec opening and 0.035 closing. This was accomplished by attaching a Marotta pilot valve (equivalent orifice size = 0.190 in.) directly to the actuator inlet port. The supply pressure corresponding to the fastest times was 800 psig GHe. Test data are shown in Figure 36.

e. Valve Disassembly and Component Inspection

Component inspection was performed at the conclusion of the initial 100 dry cycles, 5720 LN_2 cycles of the first 10,000 cycle test, and at the conclusion of each 10,000 cycle test series. All components were observed visually for apparent damage and wear. Seals and poppets were examined using a 20X microscope.

The poppet guide Armalon bearing surface indicated heavy wear on the side of the bearing adjacent to the valve inlet port. There were indications that the spring guide was being forced over to one side of the valve body by the two springs. The indication noted was graphite material rubbed into one side of the spring guide. The graphite is deposited on the body surface from wear of the piston OD seal. If the spring guide was being forced slightly to one side, then the result might have been the poppet shaft rubbing hard on one side of the Armalon bearing. Test points and inspection results are shown in Figure 37.

III, B, Cryogenic Propellant Phase (cont.)

5. Conclusions

All objectives of the OME valve program have been satisfied to the extent of design, fabrication, testing, and evaluation. The valve, however, does not meet all of the arbitrary design goals established on 30 June 1970.

Leakage past the shaft and piston seals, RACO and Delta, exceeds the 100 scc/hr maximum limit. The Kel-F compression-molded poppet did not leak during any portion of the test program. The Teflon poppet configuration would not be adequate in this application because of seat/poppet loading loss due to low temperature and material cold flow.

6. Recommendations

Additional data should be obtained on the molded Kel-F poppet and the method of seat loading with Belleville spring washers. Also to be determined would be the minimum load required to effect an adequate seal and load variation, if any, with increasing cycles and low temperature. Additional testing should also be conducted to determine the life cycle of the Kel-F poppet at cryogenic temperatures. An upper limit in excess of 100,000 cycles does not appear unreasonable from the available test data.

The poppet shaft and actuator piston Delta seals should be further evaluated. Prior to this evaluation, the design criteria for each seal application would be analyzed with respect to seal material, surface finishes of mating parts, amount of shrinkage at cryogenic temperature, velocity of moving parts, amount of load required (axial and radial) to effect a good seal, and seal material hardness at low temperature. The wedge angle of the Delta seal assembly would be changed accordingly to compensate for anticipated material property changes encountered at cryogenic temperatures.

III, B, Cryogenic Propellant Phase (cont.)

Friction tests should be performed in more detail both at ambient and low temperature. This information is important not only because of actuation requirements but also because the local temperature within the seal material reaches a point where failure or rapid deterioration results. To compensate for this, the Delta seal contact area is controllable and is a function of the individual seal design criteria.

Method of Actuation

Type	Advantages	Disadvantages	Summary
Electrical	1. No seals required 2. Repeatable timing 3. Fewer components 4. No decontamination required	1. Does not meet present engine interfaces 2. Requires more development. 3. Impact on system power requirements.	The electrical actuation was selected as the most desirable type for an improved valve. However, due to the long and expensive development required it was determined that the work could not be accomplished within the constraints of the present program. On this basis pneumatic actuation was selected but the actuator design is to be made so that electric actuation could be applied at a later date with no modification required to the valves or valve housings.
Hydraulic	1. Uses propellant, no third fluid required. 2. No limit on fluid available for actuations.	1. Pressure decay during start transient requires large area piston and large volume of fuel. Decay can also result in erratic valve timing. 2. More cavities to be decontaminated.	
Pneumatic	1. High force margin available. 2. Repeatable timing. 3. No decontamination required. 4. Meets existing interfaces. 5. Maximum utilization of SPS experience.	1. Third fluid must be carried. 2. Additional sealing required. 3. Requires least development.	

Valve Configuration

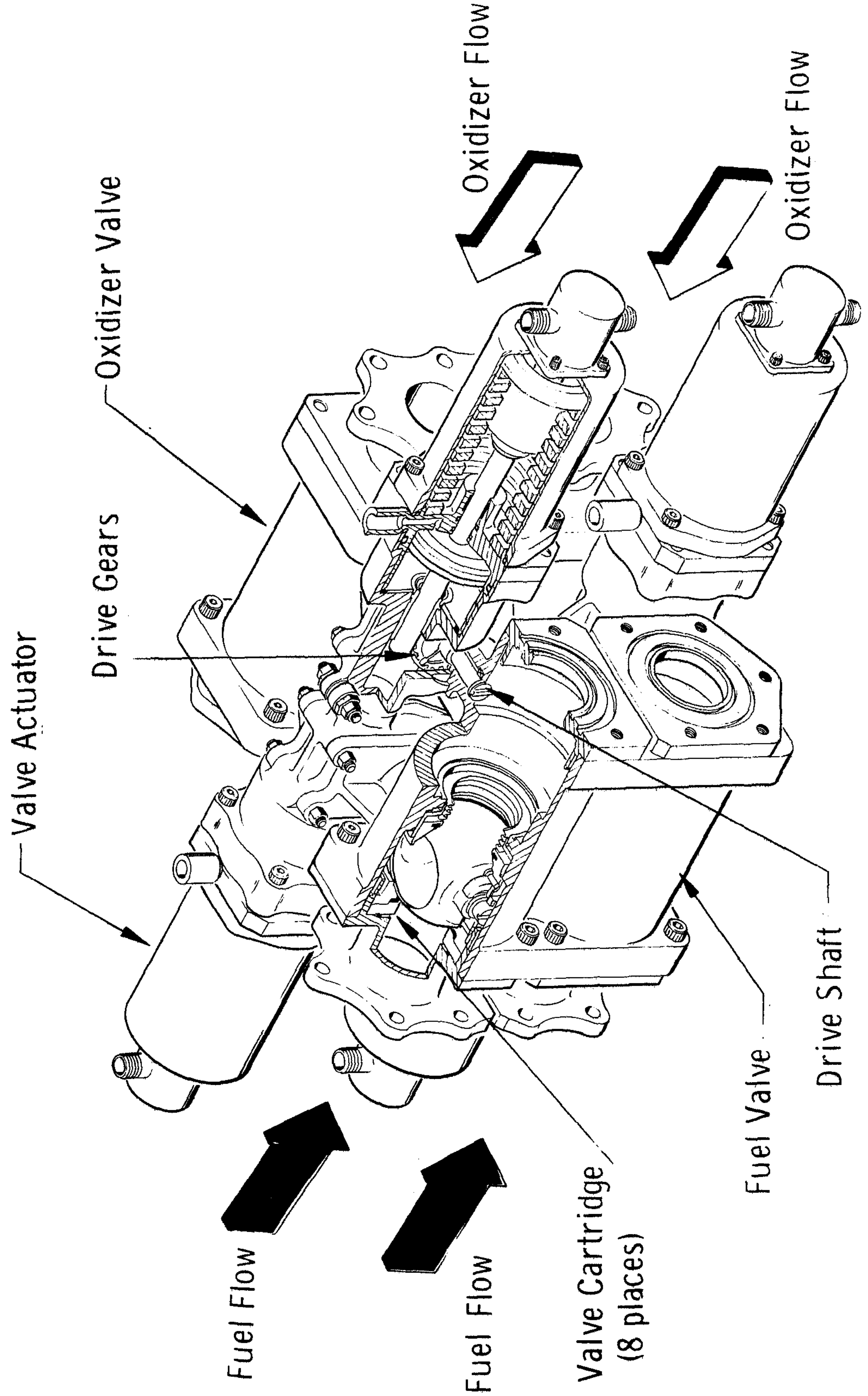
Type	Advantages	Disadvantages	Summary
Ball	1. Retains same flow characteristics as present valve. 2. Low pressure drop. 3. Makes maximum use of previous experience. 4. Can use two seals per ball.	1. Lower sealing force available. 2. Wiping motion on seal reduces seal life.	The ball type valve was selected with the primary factors in the decision being that hydraulic characteristics would be the same as the existing SPS valve and thus less engine transient testing would be required and secondly the use of the ball valve assured maximum utilization of previous experience.
Poppet			
Cam Visor Type	1. Minimum wiping of seal. 2. Low pressure drop 3. High seal loads 4. Inline flow porting for ease of redundant arrangement.	1. One seal per position. 2. High cam loads and bearing stresses. 3. Complex mechanism. 4. Different flow characteristics. 5. High actuation forces required.	
Toggle Visor	1. Minimum wiping of seal. 2. Low pressure drop. 3. High seal loads. 4. Lower compressive stresses than cam mechanism	1. One seal per position. 2. High actuation forces required. 3. Complex mechanism. 4. Different flow characteristics.	
Axial	1. High seal loads. 2. Minimum wiping of seal. 3. Least complex of poppet types.	1. High pressure drop. 2. Different flow characteristics. 3. High actuation force required.	
Blade	1. High sealing force. 2. Low pressure drop.	1. Different flow characteristics. 2. Complex mechanism required to reduce seal wiping. 3. Limited experience with this type seal. 4. Too heavy.	

Bipropellant Valve Tradeoff Chart

Figure 1, Sheet 1 of 2

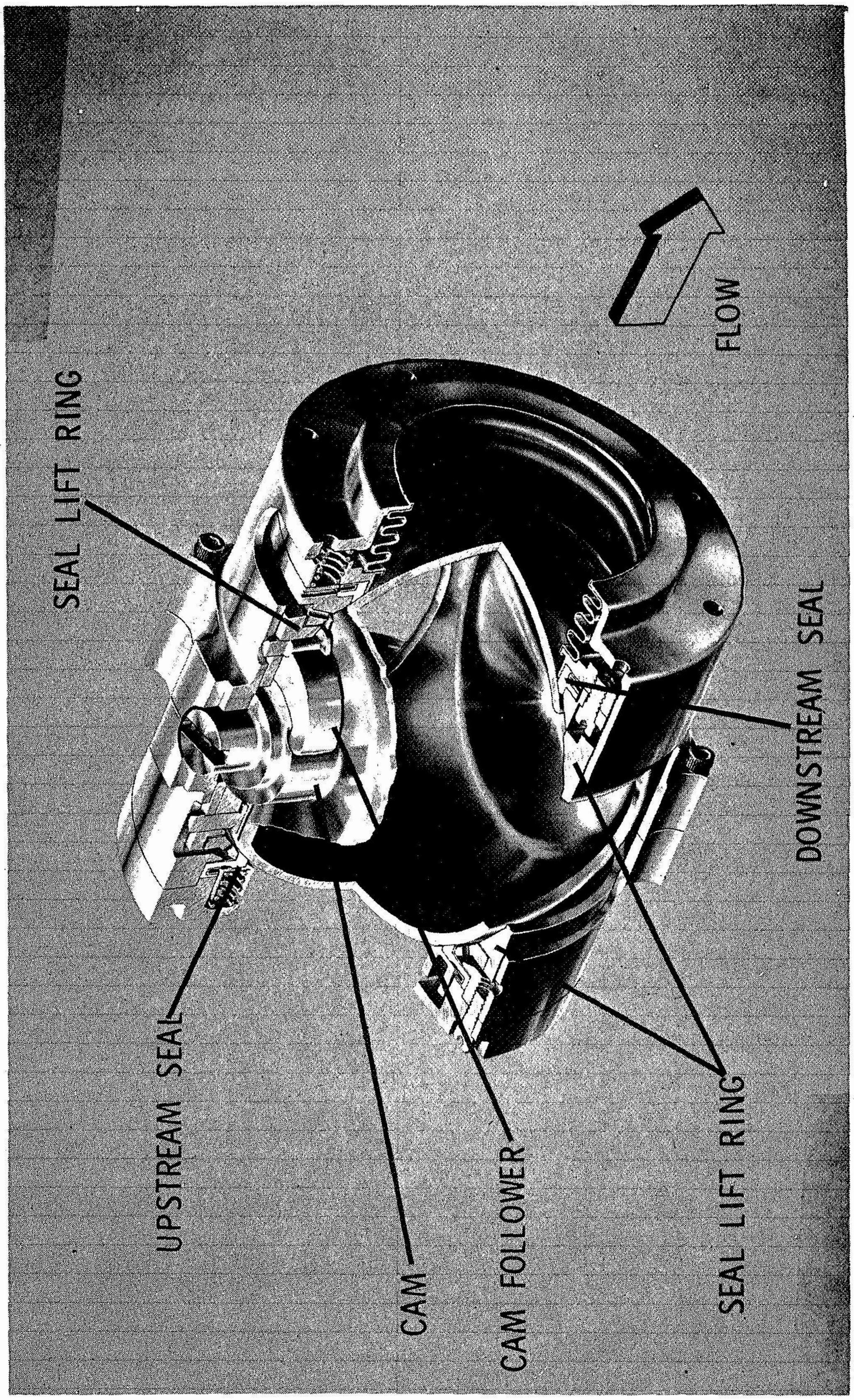
	Redundancy Configuration				Seal Configuration		
Type	Advantages	Disadvantages	Summary	Type	Advantages	Disadvantages	Summary
Quadmono-propellant Valve	1. Permits assembly/disassembly of one valve without affecting another. 2. Can be adjusted, cycled, and leak tested independent of other valves.	1. Probable increased size and weight. 2. Additional external leak paths. 3. More interface alignment problems.	The cartridge module approach was selected with the housing to be in the good monopropellant configuration. This combination provides the best accessibility to individual modules and still requires the least space and fewest external seals.	Cam Lift	1. Good potential for considerable improvement of shutoff sealing capability due to axial lift motion of seal from ball. 2. By using this concept with the ball valve configuration, maximum utilization of SPS valve technology can be realized.	1. The high cam bearing stresses and sliding action of the cam may introduce severe material and contamination problems. 2. Cam mechanism may make cartridge larger due to space required for mechanism.	The cam lift seal configuration was selected for the primary reason of least seal wiping, smallest space requirements and two seals can be used per ball.
Quadmono-propellant Module Valve	1. Any valve can be disassembled without disassembling another. 2. There are fewer internal static seals than cartridge concept. 3. Spacing between valve bores can remain same as SPS.	1. Most complex of all types. 2. Requires additional external static propellant seals for interface between the two valve and actuator units. 3. May not meet existing SPS injector and feedline interfaces.		Crank Lift	1. Good potential for considerable improvement of shutoff sealing capability due to axial lift motion of seal from ball. 2. Maximum utilization of SPS valve technology by use of ball valve. 3. Crank to link bearing stresses are low- no highly loaded metal-to-metal sliding contact is used.	1. More ball rotation required before seal is lifted away from ball than with cam lift. 2. Crank lift mechanism requires more space than mechanism.	
Single Mono-propellant Valve	1. Each module less complex than above types. 2. Can be adjusted, cycled, and leak tested independant of other valves.	1. Requires more flanges than cartridge concept. 2. Probable increased size and weight. 3. More interface alignment problems.		Eccentric Ball	1. Seal lift is accomplished with least amount of mechanical complexity. 2. Concept is adaptable to present SPS valve configuration.	1. More ball rotation required before seal is lifted away from ball than with other designs. 2. Valve positioning may be extremely critical since slight over-closure may damage seals and under-closure will not seal.	
Cartridge Module	1. Most compact of the designs. 2. Fewer flanges and static seals. 3. Each valve mechanism is identical and completely independant of one another. 4. Each cartridge will house and support both the ball and shutoff seals therefore cartridges can be assembled, cycled, and leak tested prior to assembly in the valve housing. 5. Shaft seal problems are virtually eliminated by use of universal linkage between shafts of actuator and cartridge.	1. Increased space required between the upper and lower bores and between the balls possibly requiring redesign of present injector and/or propellant feedline interfaces.					

Bipropellant Valve Tradeoff Chart



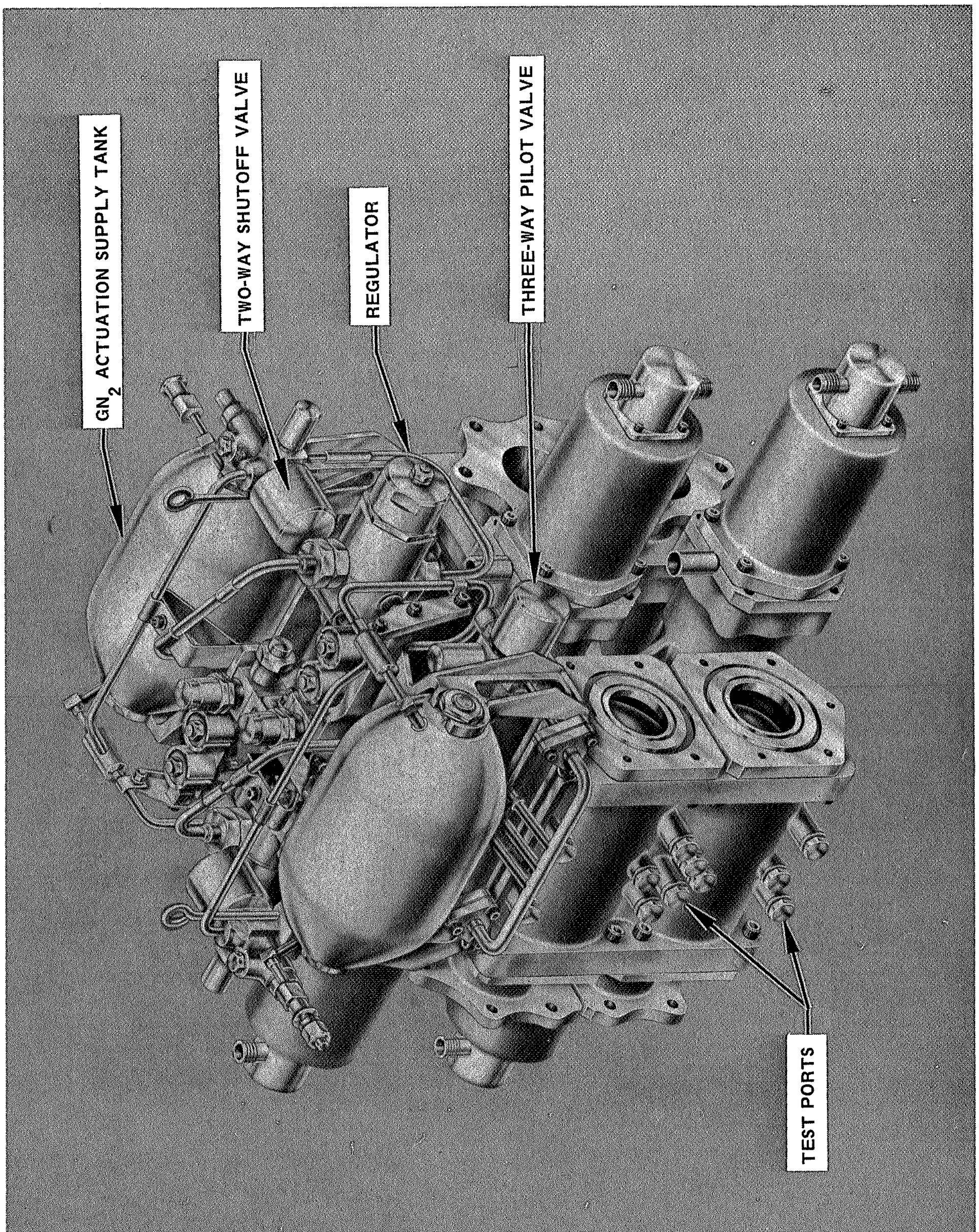
Valve Assembly

Figure 2



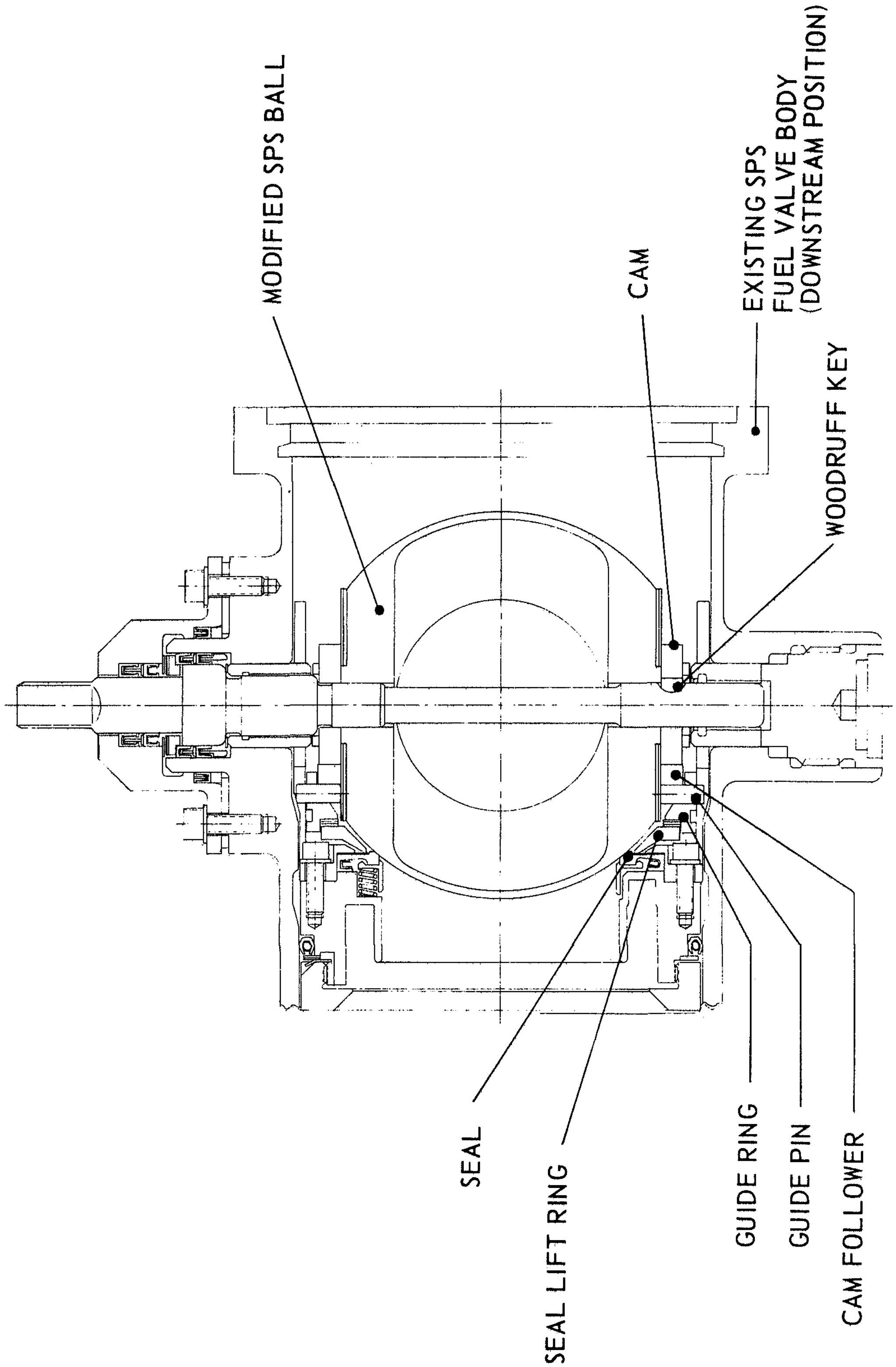
Cartridge Assembly

Figure 3



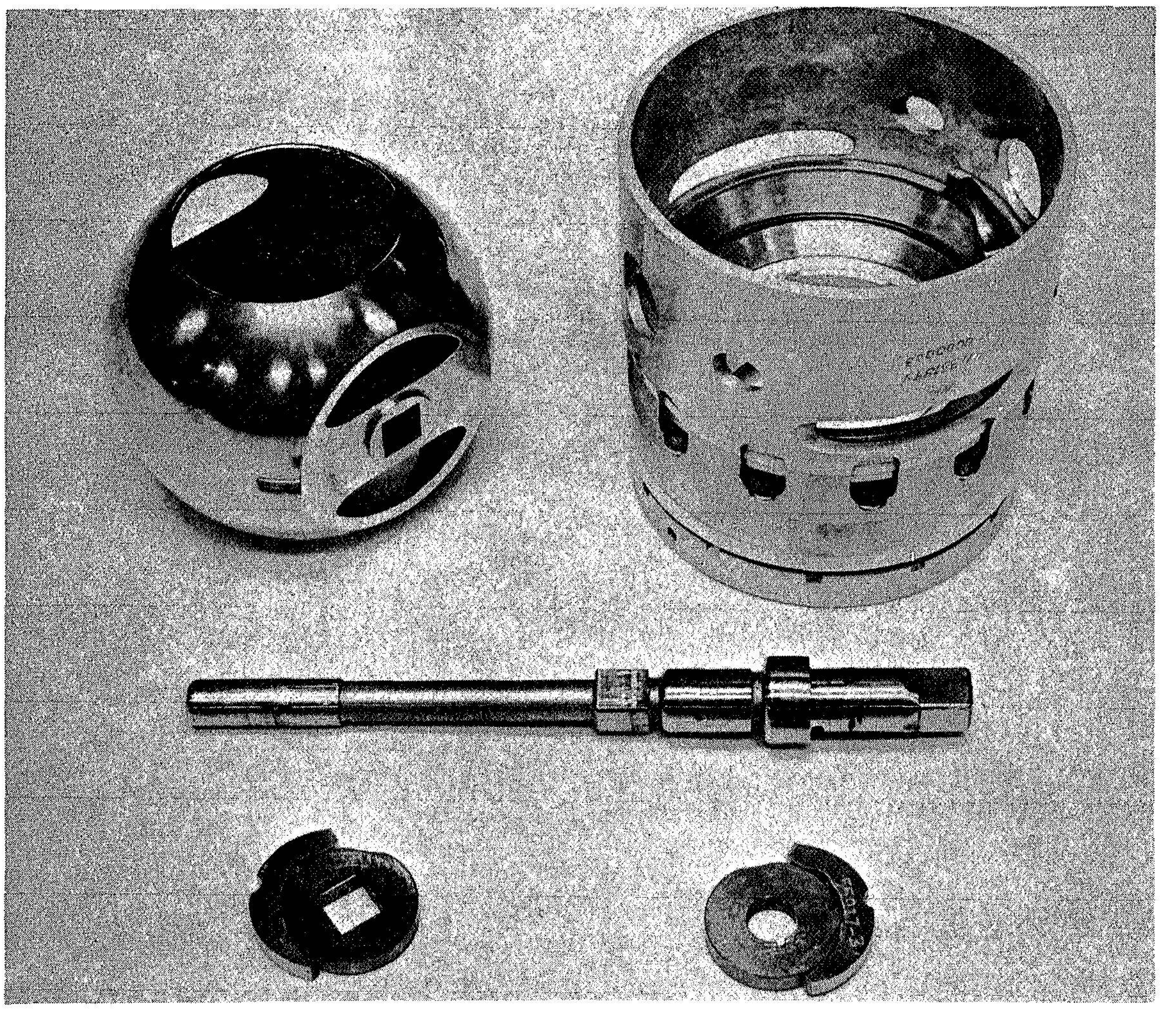
Valve and Actuator Assembly

Figure 4



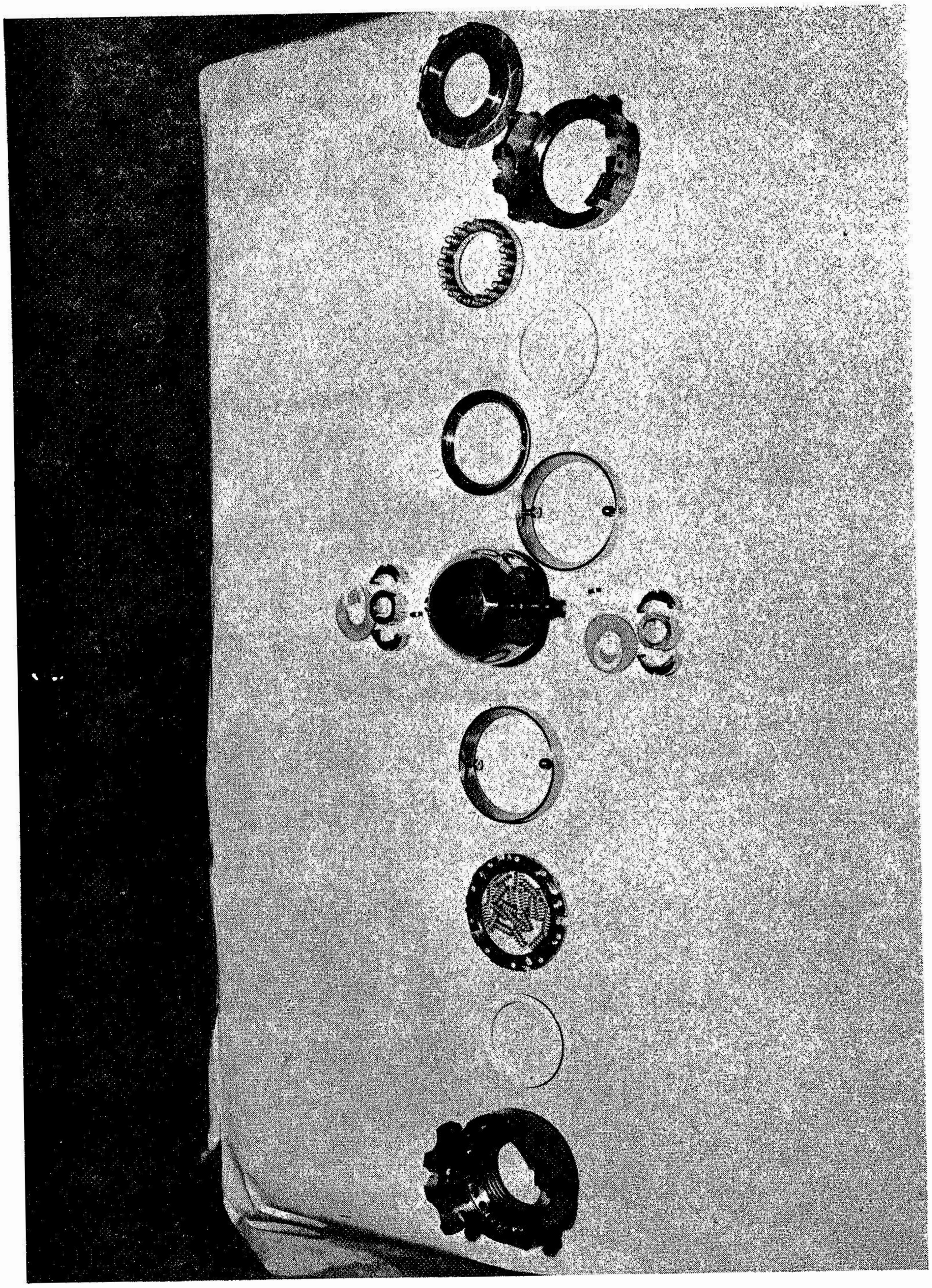
Single-Position Test Assembly

Figure 5



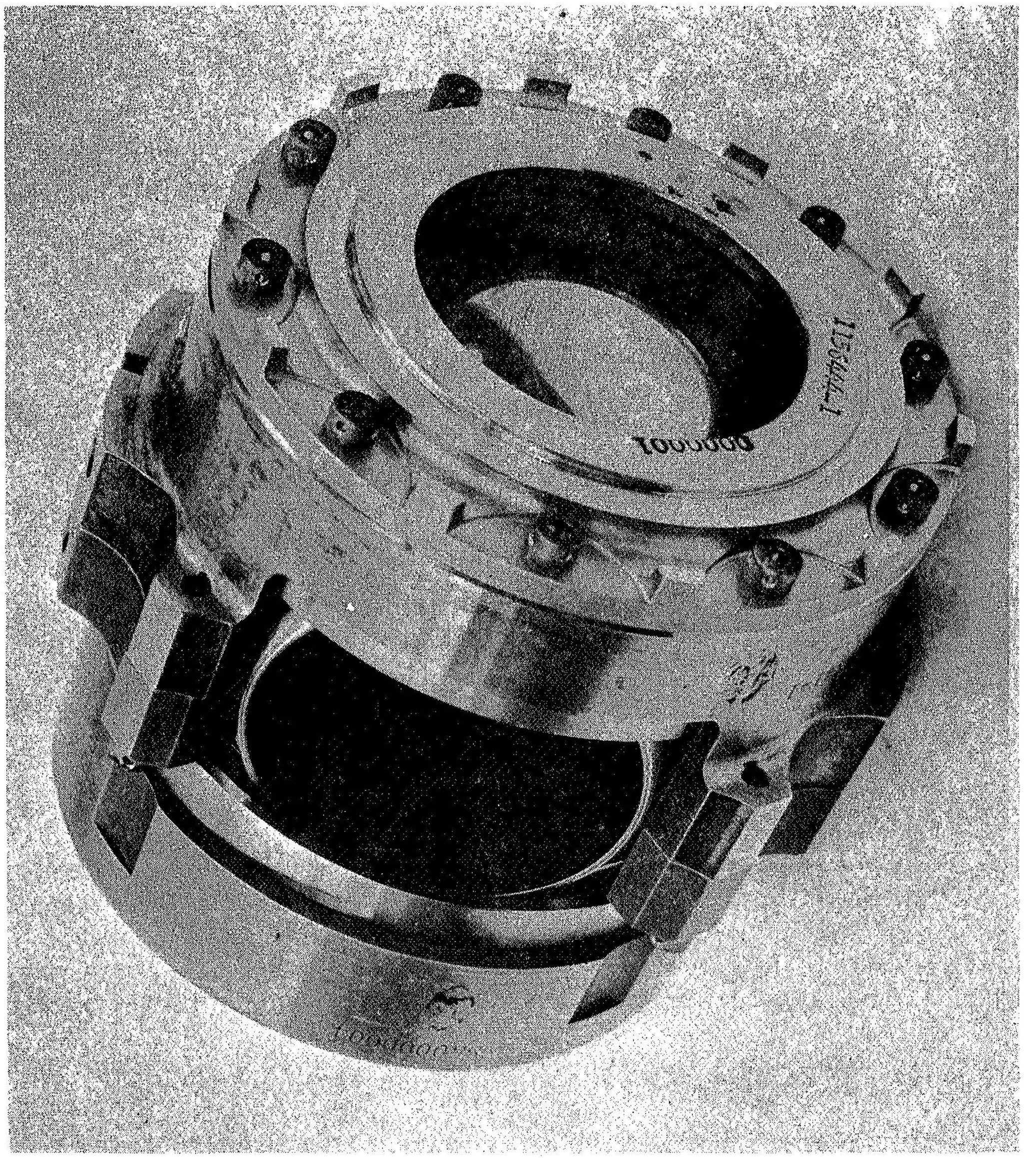
Major Components of Single-Position Test Assembly

Figure 6



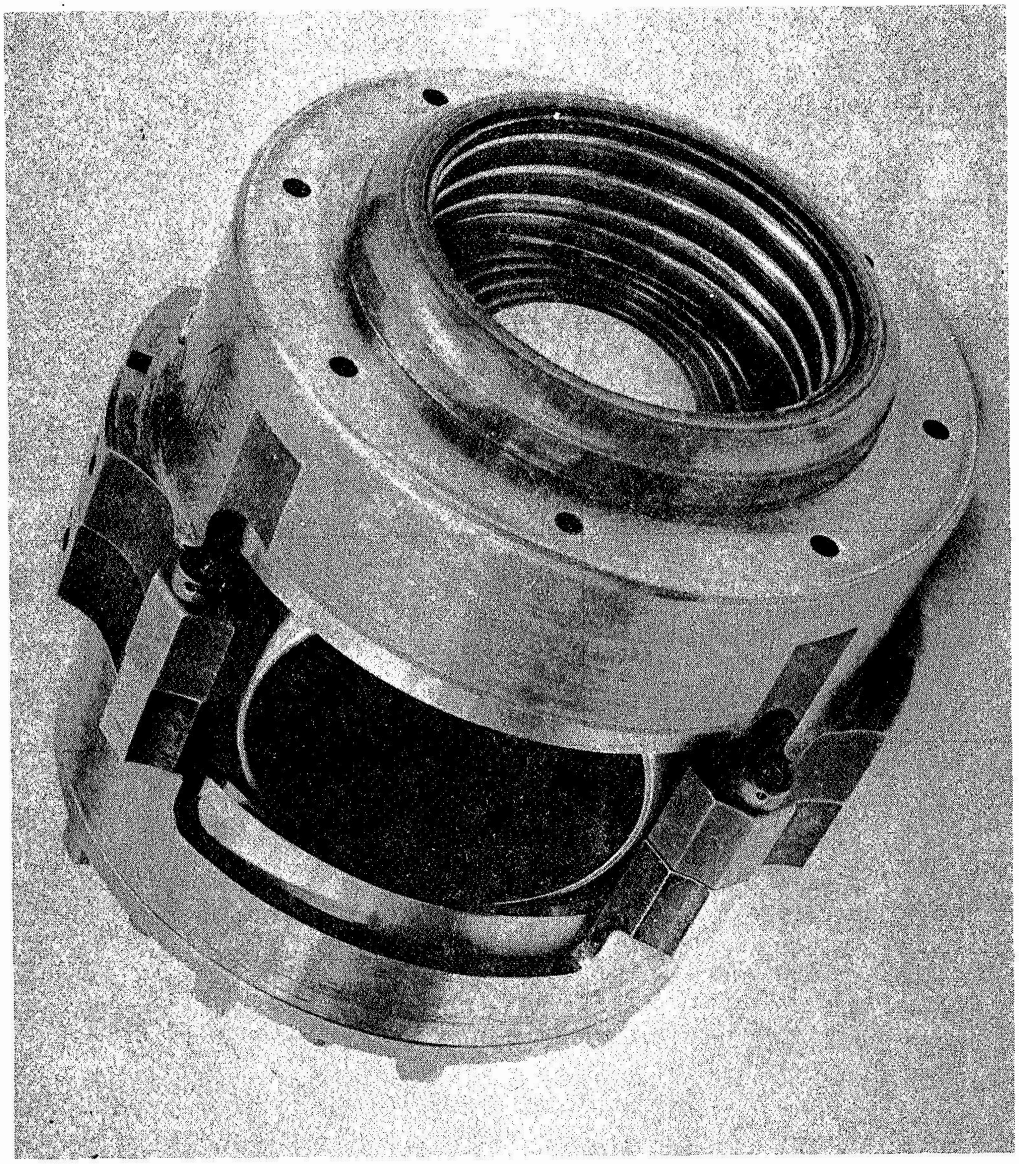
Disassembled SPS-VIP Cartridge

Figure 7



Upstream View of Assembled Cartridge

Figure 8



Downstream View of Assembled Cartridge

Figure 9

No. of Wet Cycles	Bearing and Ball Seals before Cam Lift-off Breakaway/Run	Cam Lift-Off and Opening Breakaway/Run	Valve Closing Breakaway/Run
Acceptance	25/25	48/25	30/32
64	35/30	50/35	35/32
150	45/45	55/42	42/40
250	35/32	60/42	40/45
500	45/42	60/50	45/48
1000	35/35	50/30	35/30
1500	27/27	100/85 (1)	80/75
After rework	28/28	42/30	30/28
1550	40/38	60/38	35/30
2000	35/35	70/55	50/50
2500	40/40	70/55	45/45
3500	38/38	82/60	58/58
4750	38/38	70/60	56/56
7500	35/35	65/58	52/52
10,000	30/30	65/58	52/52
15,000	42/42	58/55	54/54
20,000	40/40	60/55	55/55
20,020 (2)	35/35	80/60	62/55

(1) Cams and followers were relapped after 1500 cycles.
(2) Last 20 wet cycles of series conducted with 225-psig inlet pressure. All other cycles with 163-psig inlet pressure.

Oxidizer Valve Shaft Torque (in.-lb) - Wet Cycle Test Series

Figure 10

No. of Wet Cycles	Ball Seal No. 1					Ball Seal No. 2					Shaft Seals				
	GN_2 Leak Test Pressures, psig														
	2	75	175	240	2	2	75	175	240	2	2	75	175	240	2
Acceptance	0	0	0	–	0	0	0	16	–	0	0	0	0	–	0
64	0	0	0	–	0	0	0	0	–	0	0	0	0	–	0
150	0	0	0	–	0	96	36	10	–	0	0	0	0	–	0
250	0	0	10	–	0	0	0	0	–	0	0	0	0	–	0
500	0	70	144	–	0	0	0	10	–	0	0	0	0	–	0
1000	0	42	72	–	0	0	0	0	–	0	0	0	0	–	0
1500	0	0	0	–	0	0	0	0	–	0	0	0	0	–	0
After rework	0	18	120	–	0	0	0	60	–	0	0	0	0	–	0
1550	0	0	0	–	0	0	30	100	–	0	0	0	0	–	0
2000	0	0	88	–	10	0	30	46	–	0	0	0	0	–	0
2500	0	0	0	–	0	0	0	10	–	0	0	0	0	–	0
3000	0	0	28	–	0	0	0	0	–	0	0	0	0	–	0
4750	0	0	16	–	0	0	0	0	–	0	0	0	0	–	0
7500	0	0	0	16	0	0	0	0	16	0	0	0	0	0	0
10,000	0	0	8	0	0	0	0	0	20	0	0	0	0	0	0
15,000	0	16	110	255	0	0	0	14	16	0	0	0	0	0	0
20,000	0	120	160	208	0	0	0	0	0	0	0	0	0	0	0
20,020(1)	0	630	876	1080	0	0	0	40	28	0	0	0	0	0	0

(1) Last 20 wet cycles of series conducted with 225-psig inlet pressure. All other cycles with 163-psig inlet pressure.

Oxidizer Valve Wet Cycle Leakage Rates (cc/hr)

Figure 11

No. of Dry Cycles	Bearing & Ball Seals Before Cam Lift-Off Breakaway/Run	Cam Lift-off and Opening Breakaway/Run	Valve Closing Breakaway/Run
10	28/26	43/29	26/29
50	40/30	46/27	24/26
100	44/34	47/28	25/27
200	46/29	47/30	27/29
300	45/35	48/27	26/28
500	44/33	45/26	22/24
1000	41/31	42/25	21/24
2000	37/31	40/22	18/20
2010	44/32	42/25	22/25
3000	42/35	45/30	29/31
4000	41/32	60/45	38/40
5000	41/35	105/100	75/80

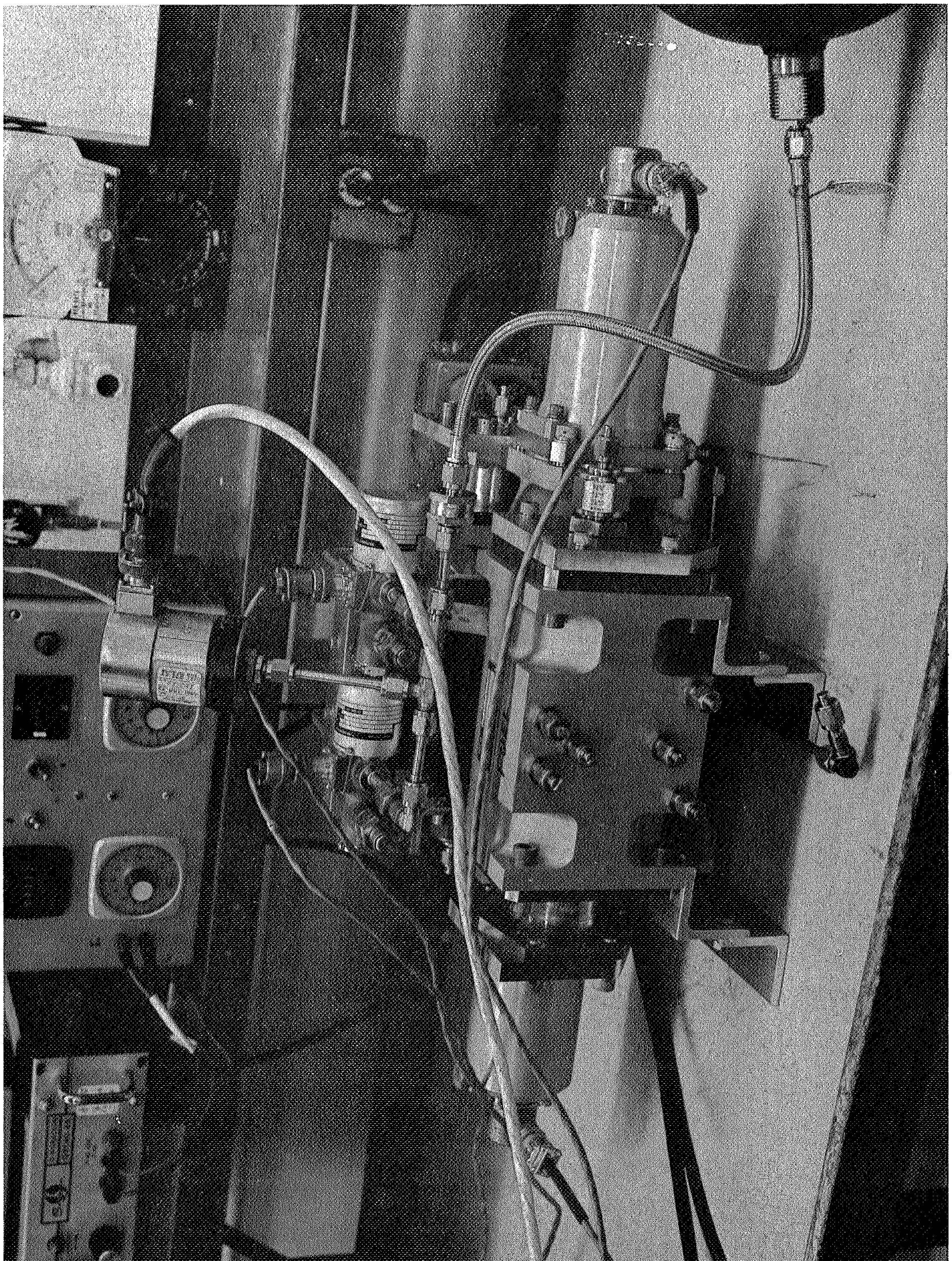
Fuel Valve SN X-2 Torque (in.-lb) Dry Cycle Test

Figure 12

No. of Dry Cycles	Ball Seal No. 1 GN$_2$ Leak 2	75	MS	2	Ball Seal No. 2 Test Pressure, psig 2	75	175	2	Shaft Seals 2	75	175	2
10	0	40	64	0	0	32	0	0	0	0	0	0
50	0	68	120	0	0	32	0	0	0	0	0	0
100	0	54	82	0	0	22	0	0	0	0	0	0
200	0	20	28	0	0	26	0	0	0	0	0	0
300	0	14	22	0	0	12	0	0	0	0	0	0
500	0	24	26	0	0	30	2	0	0	0	0	0
1000	0	16	26	0	4	44	30	0	0	0	0	0
2000	0	252	360	0	2	264	1080	2	0	0	0	0
2010	0	112	228	0	0	152	290	0	0	0	0	0
3000	6	960	1320	0	0	38	130	0	20	80	70	0
4000	90	4800	9600	62	34	1032	3600	20	280	8	0	30
5000	90	4800	10400	32	80	3660	10800	110	30	2400	3600	50

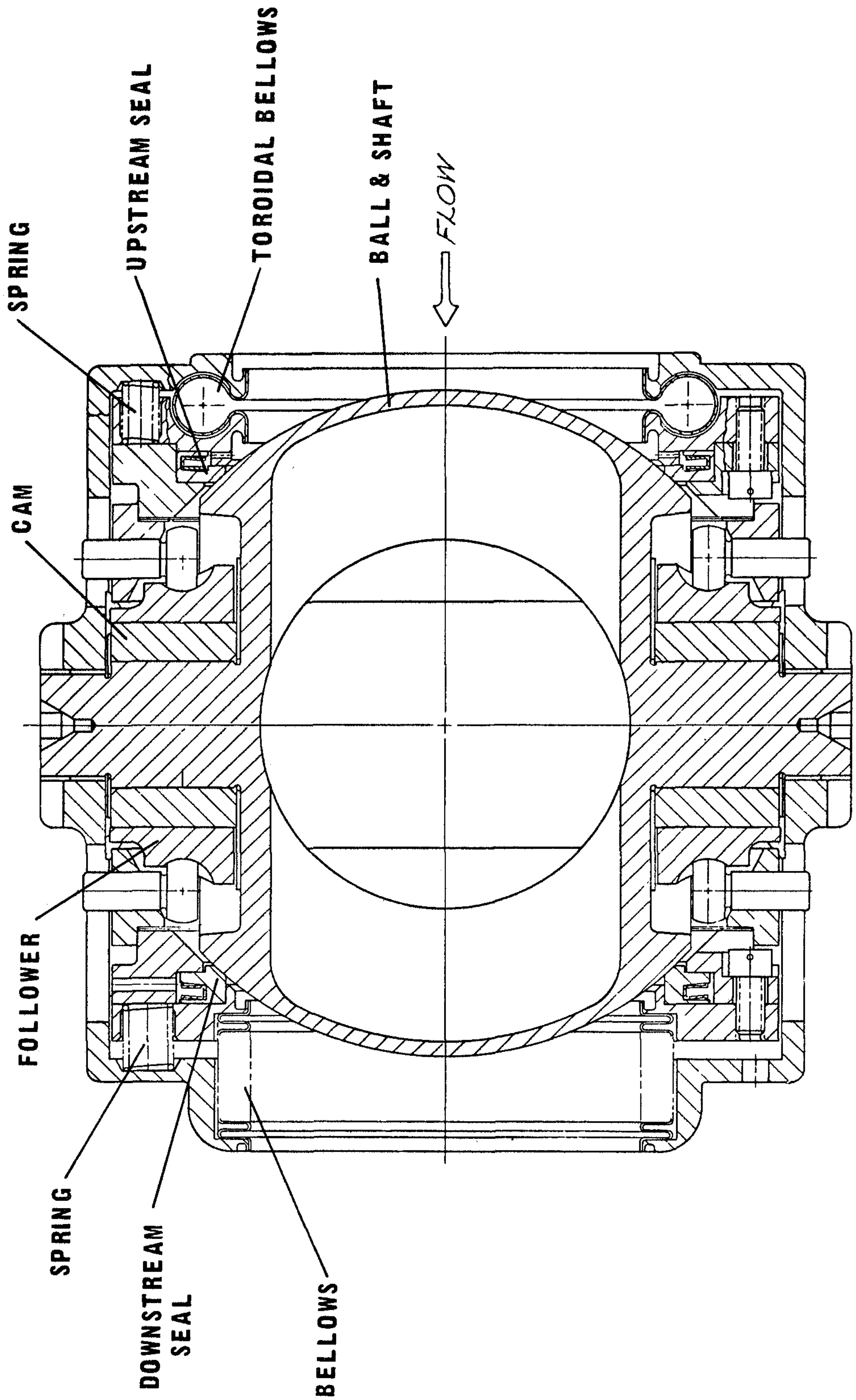
Fuel Valve SN X-2 Dry Cycle Leakage Rates (cc/hr)

Figure 13



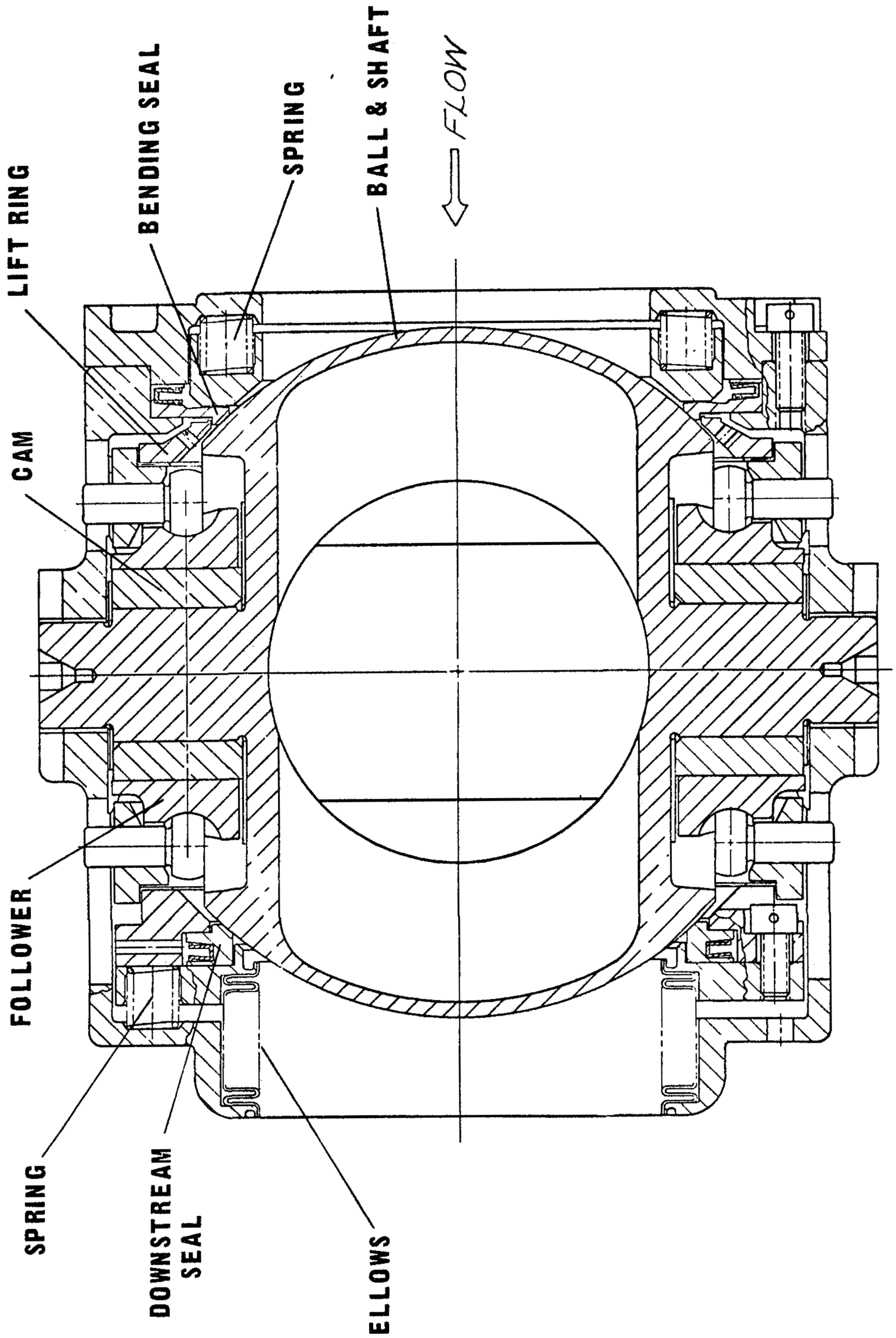
Two-Bore Valve Test Setup

Figure 14



Toroidal Bellows

Figure 15



Cartridge Assembly - Bending Seal

Figure 16

Event	Seal No. 1				Seal No. 2			Seal No. 3				Seal No. 4				Shaft Seals				
	2 (Test Pressure, psig)	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2
Acceptance Test	0	10	40	0	0	16	24	0	0	0	9	0	0	0	0	0	0	0	0	0
Post 50 Dry Cycles	0	10	30	0	0	25	45	0	0	0	10	0	0	0	0	0	0	0	0	0
Post 100 Dry Cycles	0	8	12.5	0	0	20	10	0	0	0	0	0	0	0	0	0	0	0	0	0
Post 200 Dry Cycles	0	14	30	0	0	52	52	0	0	0	0	0	0	0	0	0	0	0	0	0
Post 300 Dry Cycles	0	10	40	0	0	50	60	0	0	80	25	0	0	3	9	0	0	0	0	0
Post 500 Dry Cycles	0	0	30	0	0	35	20	0	0	0	0	0	0	0	0	0	0	0	0	0
Acceptance Test	0	0	52	0	0	0	0	0	0	10	66	0	0	10	15	0	0	0	0	0
Post 50 Dry Cycles	0	47	115	0	0	75	15	0	0	20	40	0	0	9	0	0	0	0	0	0
Post 100 Dry Cycles	0	0	0	0	0	30	0	0	0	0	5	0	0	0	0	0	95*	0	0	0
Post 200 Dry Cycles	0	8	54	0	0	10	4	0	0	0	3	0	0	0	0	0	0	0	0	0
Post 300 Dry Cycles	0	8	30	0	0	25	0	0	0	0	0	0	0	0	0	0	25*	0	0	10*
Post 500 Dry Cycles	0	0	22	0	0	5	2.5	0	0	7	12.5	0	0	0	0	0	20*	2.5*	2.5*	5*

Leakage from upstream primary shaft seals.

Fuel Valve Leakage Data (cc/Hour)

Figure 17 (Sheet 1 of 2)

Event	Seal No. 1				Seal No. 2				Seal No. 3				Seal No. 4				Shaft Seals			
	2	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2
	(Test Pressure, psig)																			
Post 250 Wet Cycles	0	210	820	96	0	220	325	13	0	52	130	11	2	1	45	0	0	0	0	0
Post 500 Wet (Oxid Only)																				
Post Fuel Half Rebuild	0	0	0	0	0	20	10	0	0	60	210	0	0	70	60	0	0	0	0	0
Post 1000 Wet Oxid - 500 Dry Fuel	0	120	360	0	0	70	92	0	0	177	312	0	0	74	36	0	0	0	0	0
Post 1500 Wet Oxid - 500 Wet Fuel	0	0	0	0	0	8	32	0	0	201	249	0	0	468	1200	0	0	0	0	0

Leakage from upstream primary shaft seals.

Fuel Valve Leakage Data (cc/Hour)

Figure 17 (Sheet 2 of 2)

Event	Seal No. 1				Seal No. 2				Seal No. 3			Seal No. 4					Shaft Seals			
	2	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2
	(Test Pressure, psig)																			
Acceptance Test	0	0	0	0	0	0	7	0	0	8	0	0	0	0	0	0	0	0	0	0
Post 50 Dry Cycles	0	0	0	0	0	0	33	0	0	17	18	0	0	0	0	0	0	0	0	
Post 100 Dry Cycles	0	0	0	0	0	8	0	0	0	120	336	0	0	12	8	0	0	0	0	0
Post 200 Dry Cycles	0	75	260	0	0	12	28	0	0	125	420	0	0	11	5	0	0	0	0	0
Post 300 Dry Cycles	0	30	150	0	0	7	21	0	0	52	102	3	0	0	1	0	0	0	0	0
Post 500 Dry Cycles	0	0	23	0	0	10	25	0	0	68	168	0	0	10	90	0	0	0	0	0
Acceptance Test	0	0	7	0	0	15	15	0	0	0	0	0	0	18	12	0	0	0	0	0
Post 50 Dry Cycles	0	37	52	0	0	25	30	0	0	25	100	0	0	8	8	0	0	0	0	0
Post 100 Dry Cycles	0	17	40	0	0	0	0	0	0	23	70	0	0	20	20	0	0	0	0	0
Post 200 Dry Cycles	0	12	35	0	0	19	27	2	0	15	33	0	0	10	15	2	0	0	0	0
Post 300 Dry Cycles	0	50	50	0	0	15	25	0	0	15	32	0	0	20	12	0	0	0	0	0
Post 500 Dry Cycles	0	25	38	0	0	16	20	0	0	22	31	0	0	25	10	0	0	0	0	0

Oxidizer Valve Leakage Data (cc/Hour)

Figure 18 (Sheet 1 of 2)

Event	Seal No. 1				Seal No. 2				Seal No. 3				Seal No. 4				Shaft Seals			
	2 (Test Pressure, psig)	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2	2	75	175	2
Post 250 Wet Cycles	0	0	0	0	0	0	1	0	0	0	7	0	0	0	0	0	0	0	0	0
Post 500 Wet Cycles (Oxid Only)	8	55	112	0	0	20	36	0	0	13	47	0	1	0	17	5	0	0	0	0
Post 1000 Wet Cycles	0	40	90	0	0	10	20	0	0	21	192	0	0	52	44	0	0	0	0	0
Post 1500 Wet Cycles	7.5	44	74	0	0	13	24	0	0	24	252	0	0	30	26	13	0	0	0	0

Oxidizer Valve Leakage Data (cc/Hour)

Figure 18 (Sheet 2 of 2)

Event	Actuator	Initial Movement	Full Open	Event	Actuator	Initial Movement	Full Open
Acceptance Test	1	56	98	Post 300 Dry Cycles	1	60	102
	2	60	100		2	63	100
Post 50 Dry Cycles	1	-	-	Post 500 Dry Cycles	1	61	104
	2	-	-		2	63	103
Post 100 Dry Cycles	1	65	104	Post 250 Wet Cycles	1	59	153
	2	68	105		2	64	145
Post 200 Dry Cycles	1	63	105	Post 500 Wet Cycles (Oxid Only)	1	77	143
	2	65	103		2	68	131
Post 300 Dry Cycles	1	53	96	Post 1000 Wet Oxid - 500 Dry Fuel	1	63	152
	2	55	93		2	65	138
Post 500 Dry Cycles	1	62	110	Post 1500 Wet Oxid - 500 Wet Fuel	1	58	173
	2	64	104		2	65	157
Acceptance Test	1	60	105				
	2	65	105				
Post 50 Dry Cycles	1	69	108				
	2	75	108				
Post 100 Dry Cycles	1	67	108				
	2	66	106				
Post 200 Dry Cycles	1	58	99				
	2	60	102				

Actuation Pressure Data (psig)

Figure 19

Event	Actuator	Opening	Closing	Event	Actuator	Opening	Closing
Acceptance Test	1	0.745	0.380	Post 300 Dry Cycles	1	0.680	0.360
	2	0.525	0.600		2	0.435	0.615
Post 50 Dry Cycles	1	0.765	0.375	Post 500 Dry Cycles	1	0.685	0.360
	2	0.525	0.600		2	0.475	0.620
Post 100 Dry Cycles	1	0.790	0.350	Post 250 Wet Cycles	1	0.816	0.378
	2	0.570	0.550		2	0.598	0.612
Post 200 Dry Cycles	1	0.722	0.380	Post 500 Wet Cycles (Oxid Only)	1	0.704	0.352
	2	0.490	0.595		2	0.468	0.582
Post 300 Dry Cycles	1	0.685	0.385	Post 1000 Wet Oxid - 500 Dry Fuel	1	0.707	0.382
	2	0.465	0.610		2	0.487	0.604
Post 500 Dry Cycles	1	0.725	0.400	Post 1500 Wet Oxid - 500 Wet Fuel	1	0.794	0.403
	2	0.508	0.590		2	0.554	0.615
Acceptance Test	1	0.710	0.365				
	2	0.510	0.620				
Post 50 Dry Cycles	1	0.705	0.350				
	2	0.505	0.590				
Post 100 Dry Cycles	1	0.730	0.360				
	2	0.520	0.605				
Post 200 Dry Cycles	1	0.690	0.350				
	2	0.480	0.600				

Valve Travel Time (sec)

Figure 20

Event	Fuel No. 1 Seal	Fuel No. 2 Seal	Fuel No. 3 Seal	Oxid No. 1 Seal	Oxid No. 2 Seal	Oxid No. 3 Seal
Acceptance Test	120	129	57	45	128	100
Post 50 Dry Cycles	115	6	55	40	124	0.5
Post 100 Dry Cycles	112	8	78	48	123	96
Post 200 Dry Cycles	104	123	52	40	131	86
Post 300 Dry Cycles	100	6	55	47	130	88
Post 500 Dry Cycles	98	17	48	49	136	84
Acceptance Test	108	100	62	79	60	55
Post 50 Dry Cycles	108	137	57	75	60	57
Post 100 Dry Cycles	105	145	54	76	60	52
Post 200 Dry Cycles	111	15	54	75	60	55
Post 300 Dry Cycles	112	115	54	70	60	53
Post 500 Dry Cycles	112	125	55	70	61	51

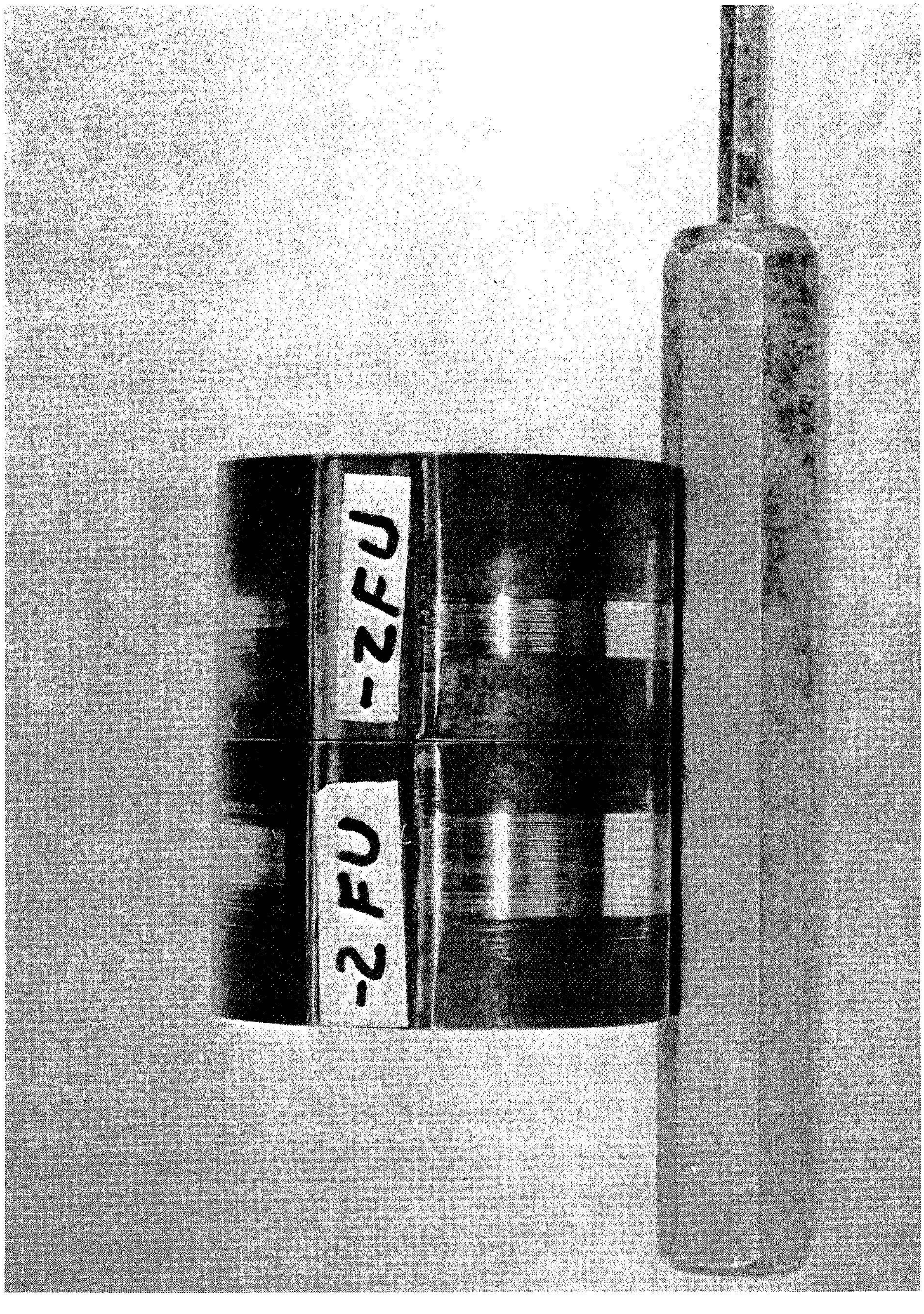
Ball Seal Back Pressure Relief Data (psig)

Figure 21 (Sheet 1 of 2)

Event	Fuel No. 1 Seal	Fuel No. 2 Seal	Fuel No. 3 Seal	Oxid No. 1 Seal	Oxid No. 2 Seal	Oxid No. 3 Seal
Post 250 Wet Cycles	95	169	41	66	43	29
Post 500 Wet Cycles (Oxid Only)	-	-	-	93	56	62
Post 1000 Wet Oxid - 500 Dry Fuel	90	50	60	87	45	27
Post 1500 Wet Oxid - 500 Wet Fuel	90	58	46	72	43	55

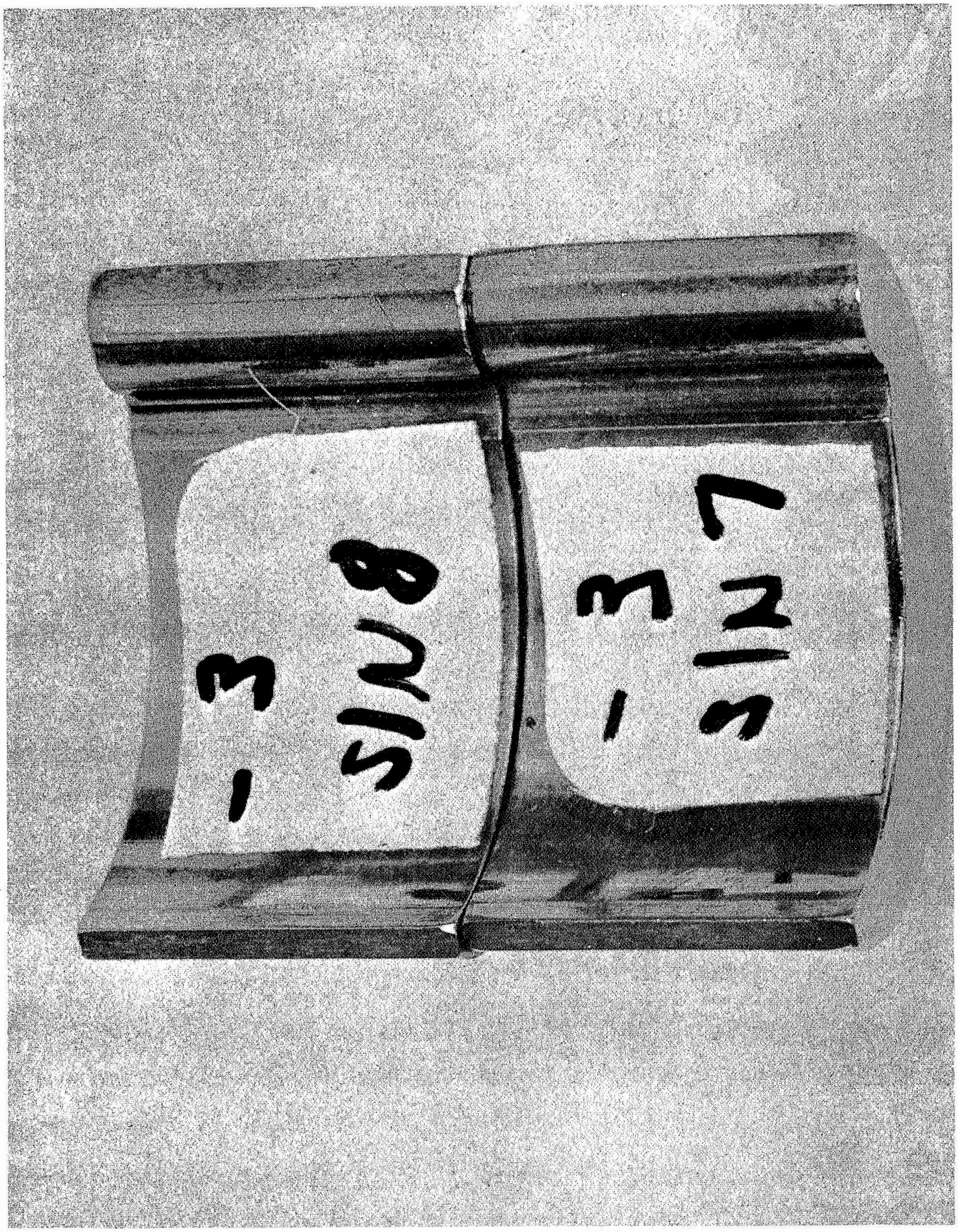
Ball Seal Back Pressure Relief Data (psig)

Figure 21 (Sheet 2 of 2)



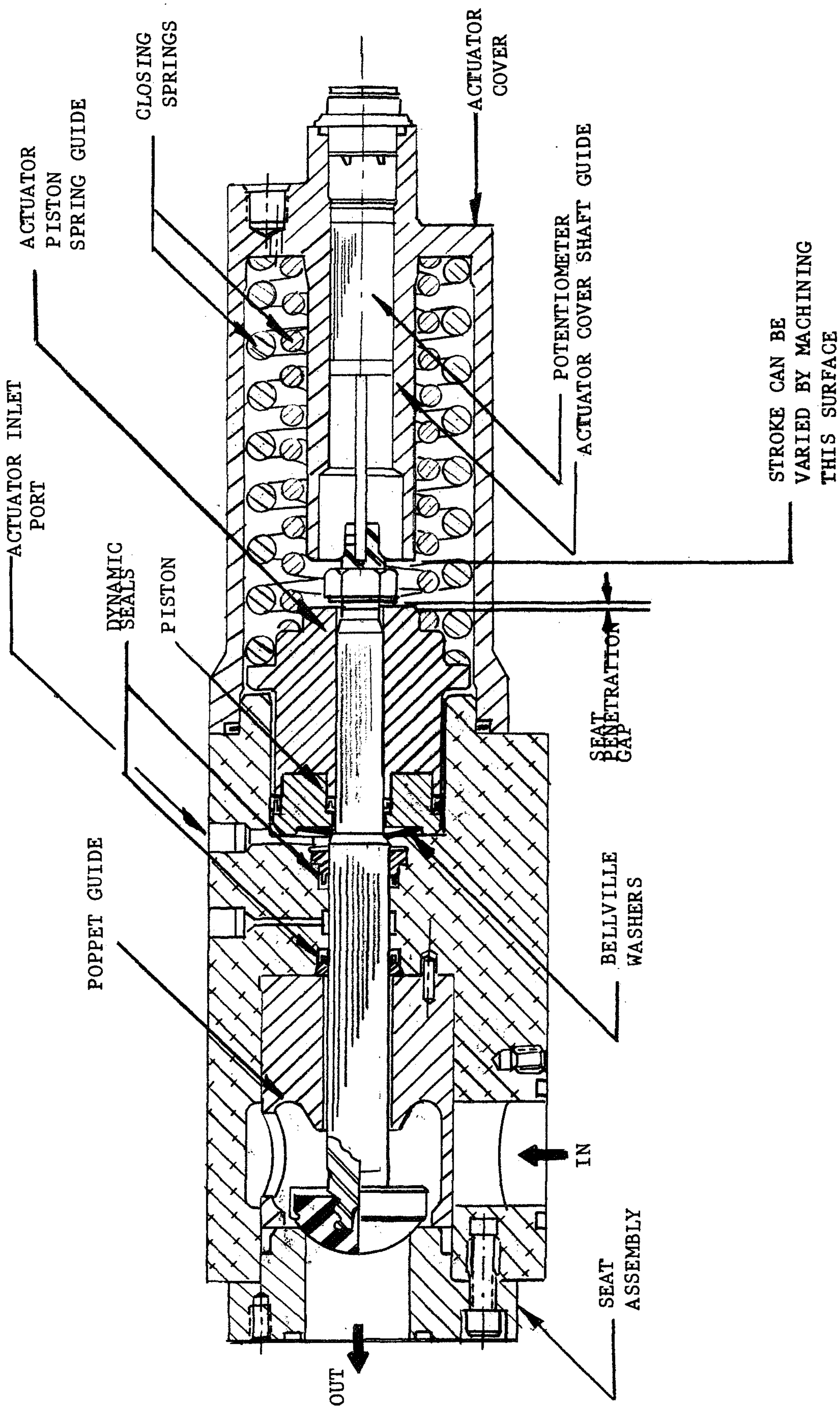
Fuel Cams after Wet Cycle Testing

Figure 22



Oxidizer Followers after Wet Cycle Testing

Figure 23



OME Valve Assembly

Figure 24

1.	Propellants	
	Fuel	Liquid hydrogen at 40°R
	Oxidizer	Liquid Oxygen at 180°R
2.	Flow Rate	
	Fuel	2.52 lb/sec
	Oxidizer	15.08 lb/sec
3.	Valve Pressure Drop	25 psi max at rated flow
4.	Valve Inlet Pressures, Test Conditions	
	Fuel	1600 psig max
	Oxidizer	1600 psig max
5.	Response Time	
	Opening	0.150 $\pm$ 0.030 sec, signal to full open
	Closing	0.100 $\pm$ 0.040 sec, signal to full closed
6.	Leakage	No bubbles in 2 minutes, initial; 100 scc/ hr max after 1000 cycles
7.	Cycle Life	1000 min
8.	Operating Temperature Range	40 to 530°R
9.	Proof Pressure	2600 psig
10.	Burst Pressure	4000 psig
11.	Other Requirements	

Valve shall fail-safe closed with loss of electrical power.

Inlets and outlets shall accommodate static flange seals.

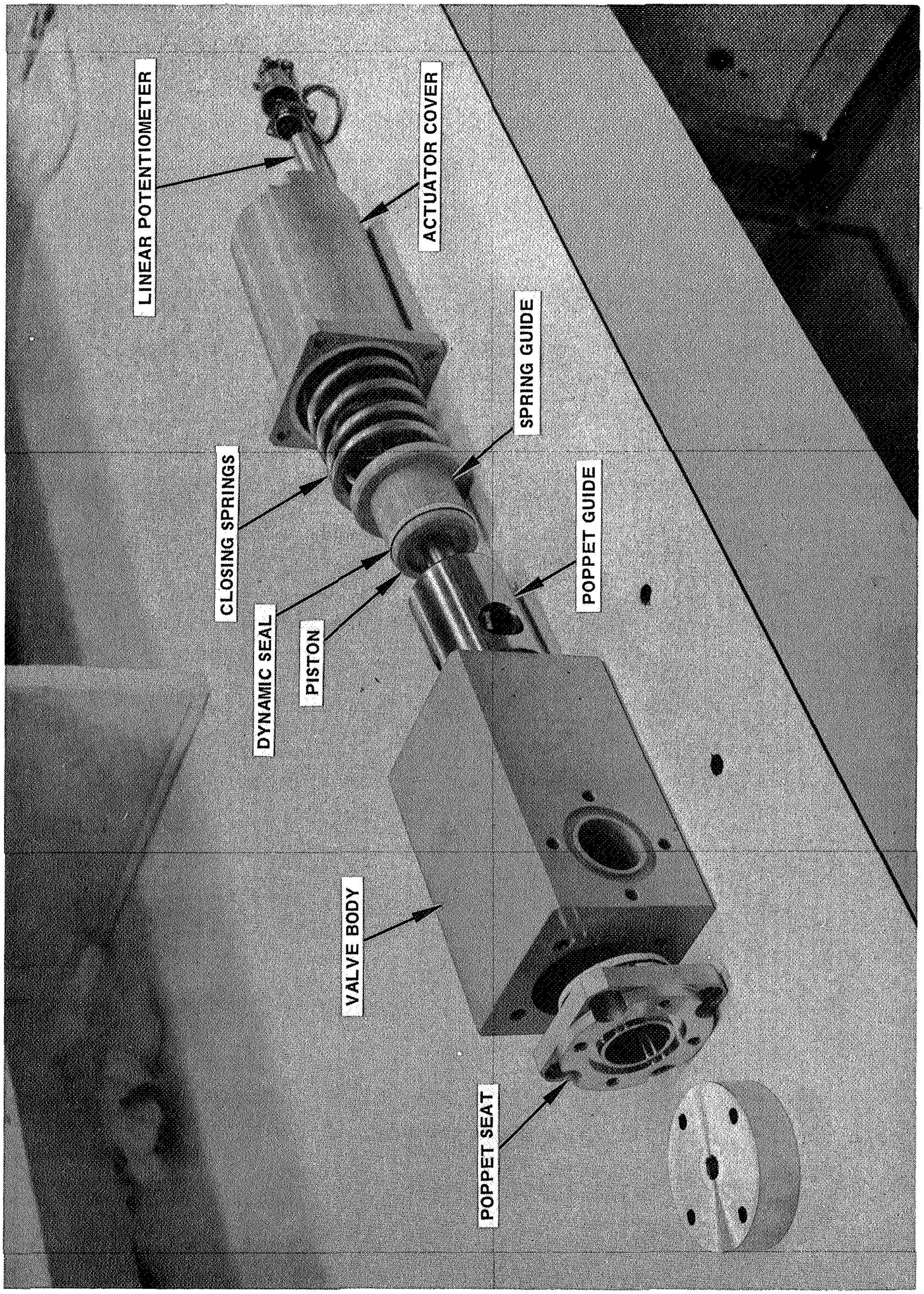
Pneumatic actuation shall be used.

Position indication shall be provided.

Valves shall be monopropellant with independent actuation.

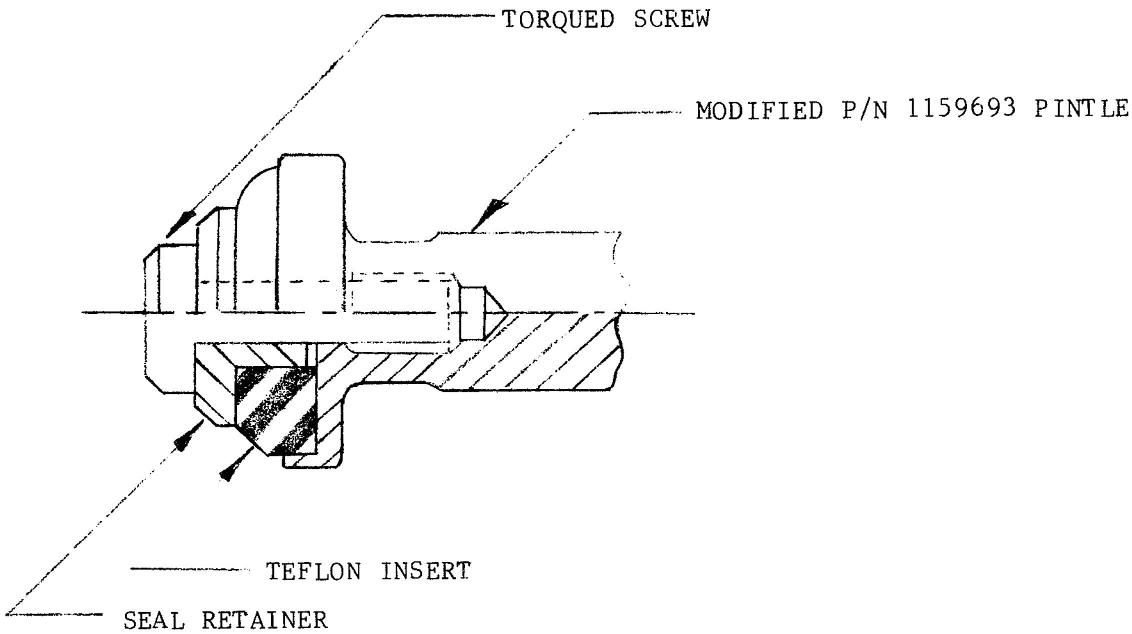
OME Valve Design Criteria

Figure 25



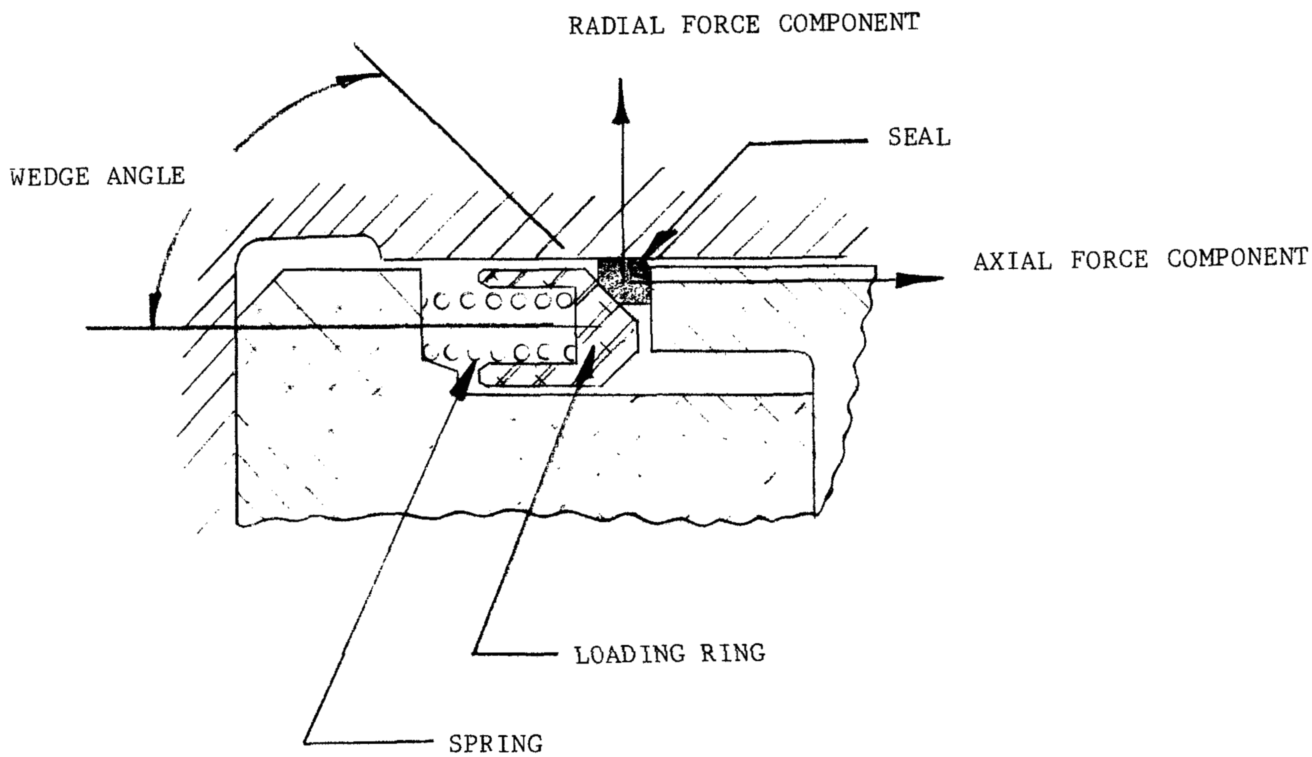
Valve and Actuator Components

Figure 26



Teflon Poppet Assembly

Figure 27



Delta Seal Installation

Figure 28

Component	PN	Quantity	Where Fabricated
Body	1159690-1	2	Associated Machine
Spacer, Seal	1159703-1	2	ALRC
Seal	10062-1-750	16	RACO
Retainer	115968-1	2	ALRC
Retaining Ring	MS16625-4112	6	United Supply
Union	MS24392J4	4	United Supply
Check Valve	2620T-4TT-5	2	James, Pond, Clark
Conical Seal	VS1015A4	10	VSI
Bolt	MS21290-12	24	United Supply
Washer	AS4012-5	12	R.G. Wallace
Seat	1159691-1	4	TECMA
Seal	10062-1-2000	8	RACO
Seal Spacer	1159703-2	2	ALRC
Poppet Assembly	1159692-1	4	Flurocarbon
Pintle	1159693-1	4	ALRC
Poppet Guide	1159694-1	2	Gross Instruments
Spring Disc	1159702-1	8	Precision Coil
Seat, Spring	1159704-2	2	ALRC
Piston	1159695-1	3	ALRC
Seal	10066-1-592	8	RACO
Seal	10062-1-1750	8	RACO
Spring Guide	1159696-1	3	ALRC
Potentiometer	421-0100	2	Beckman Instruments
Spring, Inner	1159701-1	2	Associated Spring
Spring, Outer	1159699-1	2	Precision Coil
Cover	1159697-1A	2	ALRC
Seal	10070-3250	4	RACO
Delta Seal Assembly	124385TG747	2	Rubly Engr. Co.
Delta Seal Assembly	124385TG559	1	Rubly Engr. Co.
Delta Seal Assembly	124520TG1-994	1	Rubly Engr. Co.

OME Valve Components

Figure 29

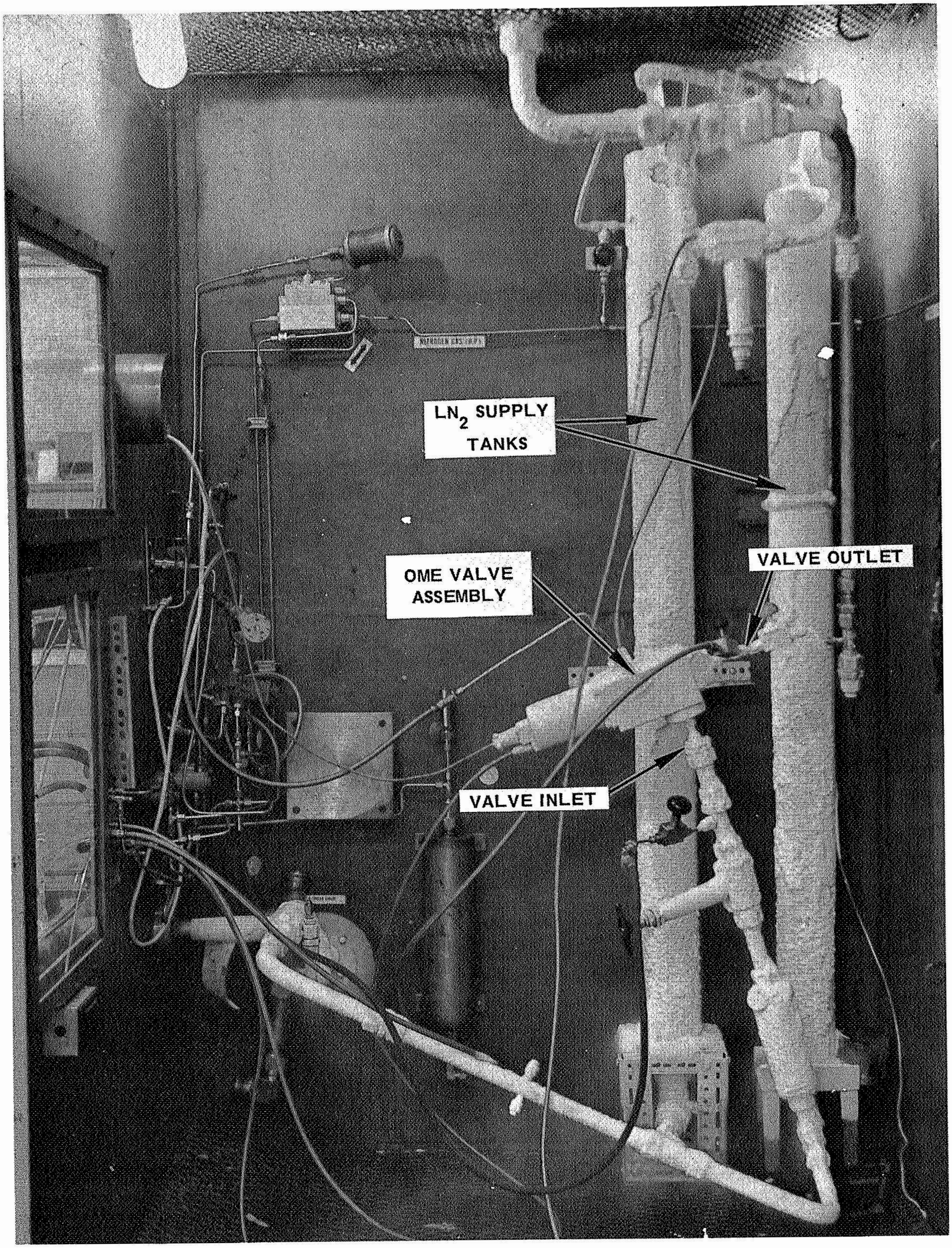
	Cycles	10	50	75	100
	Test Pressure, psig				
Poppet	10	0	0	0	0
	30	0	0	0	0
	700	0	0	0	0
Front Shaft Seal	10	0	0	0	0
	30	0	0	0	0
	700	90	60	50	60
Rear Shaft Seal	10	0	0	0	0
	30	0	0	0	0
	700	0	0	0	0
Piston Seals	10	0	0	0	0
	30	0	0	0	0
	700	8	6	3.5	3.5

NOTE: (1) Leakage rates are with helium in scc/10 min

(2) No leakage was noted from valve static seals

100 Dry Cycle Test Data

Figure 30



Low-Temperature Test Setup

Figure 31

Test	Ambient		Post 100 LN_2 Cycles		Post 500		Post 1000		Post 3000		Post 5000		Post 7000		Post 10,000		Ambient	
Pressure, psig	30	700	30	700	30	700	30	700	30	700	30	700	30	700	30	700	30	700
Piston Seals	0	35	600	2000	695	5,350	1335	16,000	160	13,350	987	1,148	0	147,000	0	37,400	0	100
Rear Shaft Seal	0	35	200	200	294	16,000	214	8,000	0	160	53	30,700	374	16,000	160	8,000	7.5	160
Front Shaft Seal	0	10	0	0	2000	2,190	1070	2,400	160	3,600	0	0	374	16,000	294	8,000	0	65
Poppet	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Actuator Cover Static Seal	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Seat Static Seal	0	0	0	Leaks	Leaking		Leaking		Leaking		Leaking		0	Leaks	0	Leaks	0	Leaks
Valve Start to Open, psig	285		285		260		280		220		280		300		290		270	
Valve Full Open, psig	330		315		340		340		300		360		340		350		325	

NOTE: Leakage rate is helium in scc/10 min

10,000 Liquid Nitrogen Cycles with Raco Dynamic Seals and Kel-F Poppet

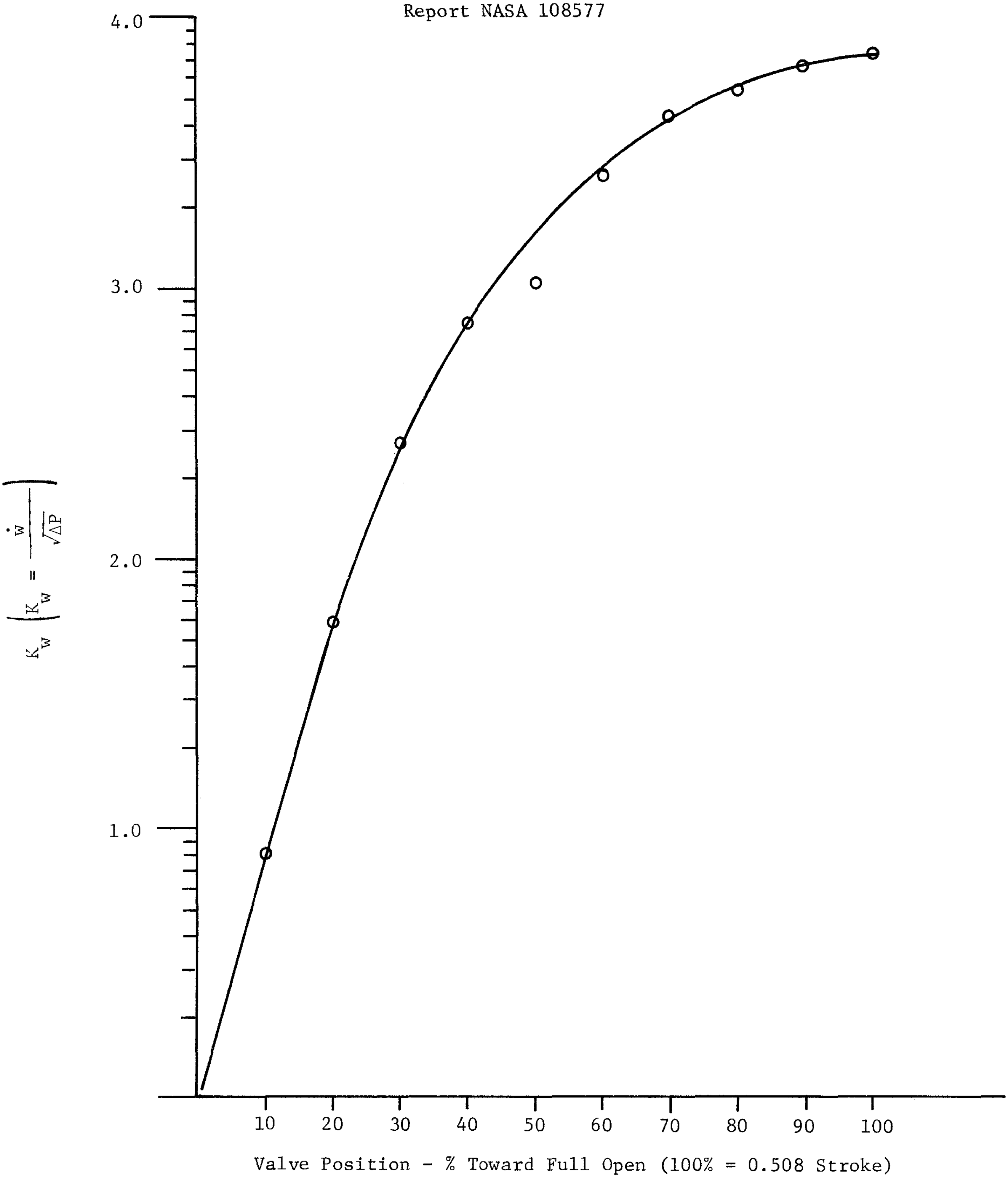
Figure 32

Test	Ambient		Post 100 LN_2 Cycles		Post 500		Post 1000		Post 3000		Post 5000		Post 7000		Post 10,000		Ambient	
Pressure, psig	30	700	30	700	30	700	30	700	30	700	30	700	30	700	30	700	30	700
Piston Seals	0	0	0	4800	0	800	0	3070	0	3600	0	3340	5	3100	0	3340	0	5
Rear Shaft Seal	0	0	25	2400	16	1740	5	667	5	320	5	1495	0	695	5	1525	0	0
Front Shaft Seal	0	0	0	200	0	65	0	160	0	54	0	15	0	36	7.5	667	0	0
Poppet	0	0	80	15	0	0	0	0	2670	0	15,000	1735	0	0	80	4	0	0
Actuator Cover Static Seal	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Seal Static Seal	0	0	0	0					Leaking									
Valve Start to Open, psig	260		--		--		290		280		290		300		290		260	
Valve Full Open, psig	290		--		--		300		340		340		350		350		310	

NOTE: Leakage rate is helium in scc/10 min

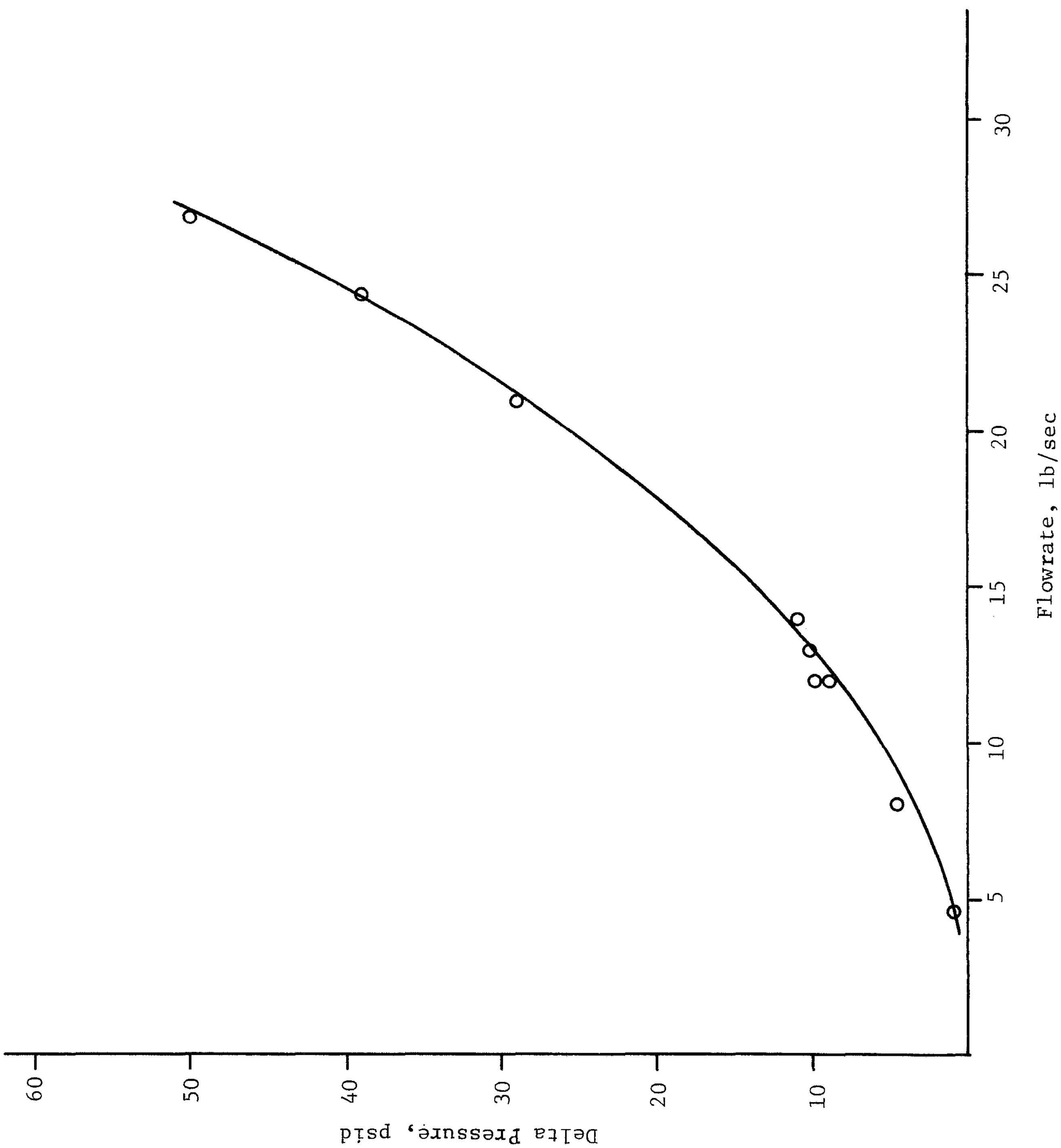
Figure 33

10,000 Liquid Nitrogen Cycles with Delta Dynamic Seals and Teflon Poppet



OME Valve K_w vs Valve Position

Figure 34



OME Valve Water Flow Pressure Drop vs Flowrate

Figure 35

	Supply Pressure, psig →	700	800	900	1000
Pilot Valve Attached Directly to Actuator Inlet	Opening Time, sec	0.009	0.007	0.010	---
	Closing Time, sec	0.037	0.035	0.040	---
Filter and 5 ft of Line between Pilot Valve and Actuator	Opening Time, sec	---	0.011	0.010	0.010
	Closing Time, sec	---	0.045	0.044	0.048

Minimum Response Data

Figure 36

POST 100 DRY CYCLE INSPECTION

Poppet	-	Very slight impression noted on seat contact sealing surface.
Actuator	-	No damage noted.
Seat Static Seal	-	A large cut noted on OD sealing surface.
Piston Seals	-	Dirt noted on sealing surface.
Shaft Seals	-	Dirt noted on sealing surfaces.
Other Metal Parts	-	No damaged noted.

POST 5720 LN_2 CYCLES (RACO SEALS AND KEL-F POPPET)

Body	-	Light deposit of spring washer material (copper) noted on surface where piston bottoms out.
Poppet	-	Teflon flake material on large OD of pintle. Teflon is from poppet guide Armalon bearing.
Poppet Guide	-	Armalon insert is wearing on one side. Worn side is toward valve inlet port. Poppet bolt torque had decreased from 80 to 35 in.-lb.
Spring Guide	-	Minor marks noted on surface where springs are positioned. This was probably caused by slight spring rotation.
Other Metal Parts	-	No damage noted.
Seals	-	No damage noted.

POST 10,000 LN_2 CYCLES (RACO SEALS AND KEL-F POPPET)

Poppet	-	No damage noted. Teflon flake material on large OD of pintle. Teflon is from poppet guide Armalon bearing.
Poppet Guide	-	Armalon insert wear noted.
Other Metal Parts	-	No damage noted.
Seals	-	No damage noted.

Disassembly Inspection Results

Figure 37 (Sheet 1 of 2)

POST 10,000 LN_2 CYCLES (DELTA SEALS AND TEFLON POPPET)

Poppet - Heavy graphite/Teflon deposits on pintle surfaces contacting dynamic seals.

Spring Guide - Delta seal material (graphite) rubbed into piston side in one location. This indicates minor contact between guide and valve body.

Poppet Guide - Teflon material tending to build up at one end due to wear from pintle. Wear did not extend to glass cloth.

Valve Body - Heavy deposits of Teflon/graphite on ID where contact is made with piston dynamic seal.

Actuator Cover - Deep scratches noted in ID where valve shaft nut is positioned. These probably occurred during assembly.

Disassembly Inspection Results

Figure 37 (Sheet 2 of 2)

STRUCTURES MATERIALS
SYSTEMS & CONTROLS DEPT.
DYNAMICS CONTROLS

VALVES AND CONTROLS SECTION

TECHNICAL REPORT NO. C.E. 5

TITLE

OME WORKHORSE VALVE TEST PLAN

CONTRACT: NAS 9-8317

PREPARED BY: J A Mullally
J A Mullally, Sr. Engineer
Controls Technology
Controls Engineering Section

APPROVED BY: J V Smith
J/V Smith, Supervisor
Controls Technology
Controls Engineering Section

DATE: 30 November 1970

OME WORKHORSE VALVE TEST PLAN

CONTRACT NAS 9-8317

I. INTRODUCTION

This test plan outlines the development test series to be conducted on the OME valve assembly. The test program is designed to evaluate the basic shutoff element and its related seals with emphasis on life cycle, leakage, and response.

Leakage past the main valve seal, shaft seals, and actuator seals will be monitored at specific intervals throughout wet or dry cycling. Actuation and timing test data will be analyzed to determine if galling, or actuator degradation, is occuring. All components will be subjected to a critical visual inspection at the completion of testing.

II. TESTS TO BE PERFORMED

Tests to be conducted are tabulated below. It is intended to conduct these tests in the order listed; any deviation from this test sequence will be documented in the applicable test request supplement. Photographs will be taken of all test setups and of any damage incurred during testing.

TEST NO.	TEST TYPE	REFERENCE PARA.	TEST LOCATION
1.	Acceptance	V A	Mechanical Controls Lab.
1.a.	Proof	V A, 1, a, b, c	" " "
1.b.	Leakage	V A, 2, a, b, c, d,	" " "
1.c.	Functional	V A, 3, a, b, c,	" " "
2	Dry Cycling	V A, 4, a,	" " "
3	Wet Cycling	V A, 5, a	" " "
4	Flow Testing	V A, 6	" " "

III. HARDWARE

One OME propellant valve, PN 1159700, will be utilized in the development evaluation. This assembly will include a workhorse actuation system supplying GN_2 to the valve actuator. An appropriate orifice will be made and installed in the workhorse system to control valve response. All assembly, testing, and disassembly will be accomplished in the Mechanical Controls Laboratory, Building 2002. Figure 1 presents a cross section of the OME propellant valve.

IV. GENERAL TEST CONDITIONS

Cleanliness of all parts, subassemblies, and the valve assembly will be maintained equivalent to AGC-46652, Level E; however, they will not be certified as such. All assembly work will be accomplished in laminar flow benches by personnel wearing nylon gloves. All parts will be sealed in clean nylon bags until required for assembly.

Test fluids will be filtered through 5 micron nominal, 18 micron absolute, filters prior to entering the valve. Valve assembly service ports which must be vented to atmosphere during testing will be protected by 5 micron nominal, 18 micron absolute, filters. All valve ports will be appropriately covered when the valve is not in use.

V. DESCRIPTION OF TESTS

A. Acceptance Test

The acceptance test will consist of a proof test, functional test, and leak test as follows:

V, Description of Tests (continued)

1. Proof Test

All proof pressure tests will be performed in a safe, remote area with gaseous nitrogen as the test fluid. No evidence of permanent distortion or damage will be acceptable. Pressure applied to more than one port at a time will be increased and decreased simultaneously at all ports.

a. Valve Assembly Proof Test

The inlet port, outlet port, actuation system supply port, and shaft seal drain port will be simultaneously pressurized to 1400 $\pm$ 10 psig for five minutes minimum. The actuation system inlet port shall be left open during the proof test.

b. Valve Seal Proof Test

The inlet port shall be pressurized to 1400 $\pm$ 10 psig for five minutes minimum. The outlet port, shaft seal drain port, and the actuation system supply port will be left open during the proof test.

c. Actuation System Proof Test

The actuation system inlet port shall be pressurized to 2000 $\pm$ 20 psig for five minutes minimum. The seal cavity drain port will be left open during the proof test.

2. <u>Valve Leak Check</u>

All ambient leakage tests will be conducted using gaseous nitrogen. Internal leakage shall be measured using water displacement type meter. External leakage will be checked by use of liquid leak detector. Leakage tests are specified for a duration of 3 ± 0.5 minutes. If leakage is observed during that period, then the duration shall be extended to 10 ± 0.5 minutes.

a. <u>Valve Shutoff Seal and Shaft Seal Leak Check</u>

The inlet port shall be pressurized with gaseous nitrogen to 10 ± 0.5 psig for 3 ± 0.5 minutes. Leakage shall be monitored from the valve outlet port and shaft seal cavity drain port. The test shall be repeated at 30, 100, 200, 500, and 700 psig.

b. <u>Actuation System Leak Check</u>

The actuation system supply port shall be pressurized to 10 ± 0.5 psig for 3 ± 0.5 minutes. Leakage past the shaft and piston seals shall be monitored from the seal cavity drain port and actuator cover vent check valve port. The test shall be repeated at 1000 ± 10 psig.

c. <u>Valve External Leak Check</u>

The inlet and outlet ports shall be pressurized to 10 ± 0.5 psig for 3 ± 0.5 minutes. Leakage shall be monitored past the valve seat static seal and inlet/outlet flange seals. The test shall be repeated at 30 and 700 psig.

2. Valve Leak Check (continued)

d. Actuator Cover External Leak Check

Pressurize the actuator cover and actuator supply port to 10 $\pm$ 0.5 psig for 3 $\pm$ 0.5 minutes. Leakage shall be monitored at the cover to valve body static seal, actuator inlet fitting static seal, and potentiometer to cover static seal.

e. Shaft Seal Cavity Drain Fitting Static Seal Leak Check

Pressurize the valve inlet port, seal cavity drain port, and actuator supply port to 10 $\pm$ 0.5 psig. Leakage shall be monitored past the shaft seal cavity fitting static seal. The test shall be repeated at 30 and 700 psig.

3. Valve Functional Test

The functional test will be performed in three segments, an actuation test, timing test, and maximum response capability. Normal post cycling tests will only include the actuation and timing tests. The maximum response capability test shall be performed at the conclusion of the test program.

3. Valve Function Test (continued)

a. Actuation Test

The intent of this test is to determine changes in the breakaway friction of the valve assembly. Gaseous nitrogen shall be applied slowly to the actuation supply port until the piston moves to the open position as verified by the potentiometer trace. Pressures corresponding to initial piston movement and full travel shall be measured and recorded. The data shall be analyzed to determine that no large change in friction has occurred.

b. Timing Test

This test will verify the proper selection of timing orifices during test assembly and will identify valve timing changes from test to test. The test fluid will be gaseous nitrogen. The actuation system inlet pressure, the energization and de-energization signals, and the potentiometer outputs will be measured and recorded.

For each timing test conducted, the actual opening and closing times will be recorded. These data will be compared with previous tests to determine if significant changes have occurred.

Required total valve response times are .150/.170 seconds for both opening and closing.

3. Valve Function Test (continued)

c. Minimum Response Testing

At the conclusion of the test program, the fastest response capability of the valve shall be determined. This test will be based on the designed actuator piston operating pressure and appropriate orifice changes. Main shutoff seal and actuator component degradation, if any, shall be observed during this testing.

4. Valve Dry Cycle Testing

a. The valve shall be cycled 100 times, at the required response time, with 30 psig of gaseous nitrogen supplied to the inlet port. A minimum of two seconds open and three seconds closed shall be required. Intermittent leak testing shall be conducted following 10, 50, 75, and 100 cycles as follows:

a.1 Main shutoff seal leak checks at 10, 30, and 700 psig.

a.2 Shaft seal leak check at 10, 30, and 700 psig.

a.3 Actuator piston and shaft seal leak check at required actuation system pressure.

a.4 A functional and timing test shall be performed only at the end of 100 dry cycles.

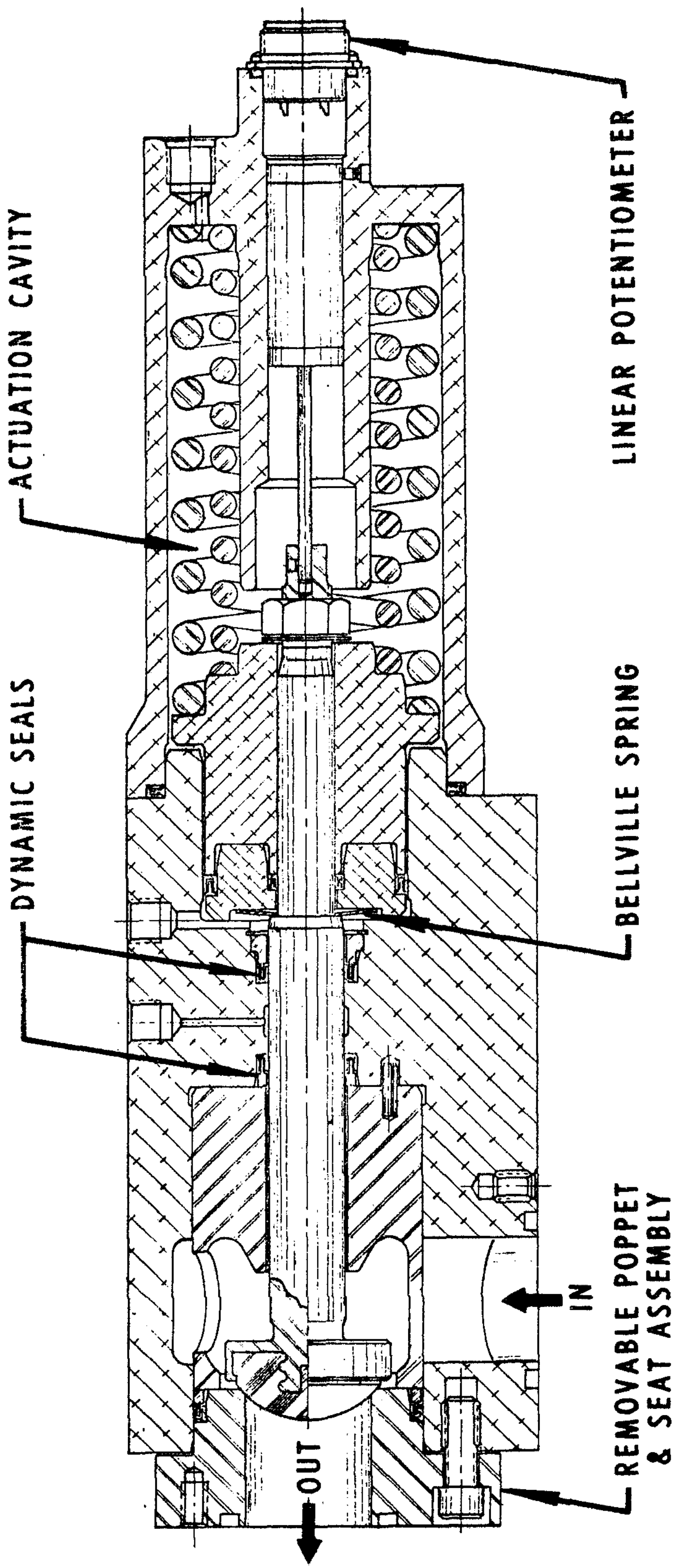
5. <u>Valve Wet Cycle Testing</u>

The wet cycle test will consist of cycling the valve with liquid nitrogen applied to the valve inlet at 700 $\pm$ 10 psig. The valve shall be cycled a total of 10,000 times. Liquid nitrogen shall be supplied to the valve through the 1/4 inch inlet boss on the valve inlet closure plate. The required response time shall be maintained throughout the cycling test. A minimum of one second open and three seconds closed shall be required. Intermittent tests shall be performed at 100, 500, 1000, 3000, 5000, 7000, and 10,000 cycles as follows: (Gaseous helium shall be used during all leakage tests)

a. Intermittent testing shall consist of main shutoff seal leak checks at 30 and 700 psig, shaft seal leak checks at 30 and 700 psig, actuator piston and shaft seal leak checks at 10 psig, and at the required actuation pressure, and valve functional and timing tests.

6. <u>Valve Water Flow Testing</u>

Valve water flow testing shall be conducted in order to determine the pressure drop, flow rate, and resistance coefficient (K_w) at the nominal operating conditions. These characteristics of the valve will be measured in the full open position and selected increments of valve closure.



AJ-8 Workhorse Valve Assembly

Figure 1